Cranfield University

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Assessment of Cool Thermal Storage Strategies in Kuwait

School of Engineering

PhD Thesis

Cranfield University

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Ph.D. THESIS

Academic Year 2006-7

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Academic Year 2007

ABSTRACT

The extensive use of air conditioning for indoor cooling in offices and large commercial buildings in Kuwait represents a major part of the power and electricity consumption in that country. The main objective of this research was to investigate ice and chilled water cool thermal storage technologies and operating strategies for air conditioning. This was motivated by the extreme climatic conditions in Kuwait and the necessity to reduce both maximum power demand and energy consumption whilst being economically feasible. This work represents the first such study undertaken.

In Kuwait approximately 45% of the total annual exported electrical energy is consumed solely by air conditioning systems as a result of the very high ambient temperatures occurring between March and October. Furthermore, it was estimated air conditioning systems represent about 62% of the peak electrical load.

To assess the potential of cool thermal storage, the air conditioning system for an existing clinic building, representing a typical medium size building in Kuwait, was designed with and without a cool thermal storage system. The results demonstrate that internal ice-on-coil and chilled water storage systems are suitable storage technologies that can be implemented in Kuwait.

The cooling demand of the clinic building was first estimated using the ESP-r building energy simulation program, following which the different components in the air conditioning systems were sized including chiller, storage tanks, pumps, air handling units for conventional, ice and chilled water storage air conditioning systems operating with load levelling, 50% demand limiting and full storage strategies. The heat gains by different auxiliary components in the air conditioning systems were estimated and the final cooling demand profiles were developed.

For each air conditioning design, the power and energy consumption for the design day condition and over the whole year were calculated and analysed. Furthermore, the life cycle costs were determined based on the estimated capital, maintenance, operating costs and a financial analysis was carried out.

For the Kuwaiti climate, the results demonstrate ice and chilled water storage systems can reduced the maximum power consumption during the day time when the

electricity demand is high and largest reduction in the maximum power achieved full storage strategy. However, the energy consumption of ice storage system operating with 50% demand limiting and full strategies were found were found to be higher than the conventional air conditioning system. Nevertheless, the energy consumption in the ice storage system with a load levelling operating strategy was slightly lower. Chilled water storage system was found to be unlike ice storage system, the energy consumption in all operating strategies improved over the conventional system.

Based on the estimated life cycle cost using the actual operating costs for both the government and user, it was established that for the government, ice storage operating with load levelling strategy and all other strategies of the chilled water storage systems would be more economical than conventional systems. However, for the user, load levelling ice storage, load levelling chilled water storage, and 50% demand limiting chilled water storage systems would be more cost effective.

Out of all alternatives, chilled water storage system with a load levelling strategy was found to be the most cost effective for the climate of Kuwait and for similar climates of Kuwait. Although, the outcome from this research work can not be generalised however, the method of sizing and energy and economic analysis, which was discussed in this thesis can be generalised and followed to evaluate the impact of cool thermal storage systems on energy performance and economy of the air conditioning systems.

ACKNOWLEDGEMENTS

I wish to express my sincere thanks and gratitude to my supervisor Dr. Philip Rubini for his support, valuable and continuous advice, guidance, thoughtfulness and assistance throughout the duration of this work.

My thanks also go to Kuwait Institute for Scientific Research (KISR) for the provision of the scholarship and financial support for the duration of my study at Cranfield University.

The support and understanding of Dr Essam Assem, the manager of the Buildings and Energy Technology (BET) department at KISR has been highly appreciated. Special thanks are due for my colleague and dear friends Dr. Abdulatif Ben-Nikhi and Mr. Gopal Maheshwari for their thoughtfulness, advice and support.

My gratitude belongs to my mother and all my family, whose love, encouragement, support and faith meant a very great deal to me. I would like to express my appreciation for all the kind support they have offered. It would have been much harder without such support.

Thanks to my lovely kids Maryam, Ali, Sara and Ammar.

Finally, but no means least, I would like to express my gratefulness and gratitude to my wife Bashayer for her unconditional support, encouragement and tolerance over the life of this PhD. Without her, it would simply not have been possible.

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ABBREVIATIONS

AC	Air-conditioning
ARI	Air-conditioning Refrigeration Institute
ASHRAE	American society for Heating Ventilation Refrigeration and Air-conditioning Engineer
COP	Coefficient Of Performance
KISR	Kuwait Institute for Scientific Research
MEW	Ministry of Electricity and Water
MOE	Ministry Of Energy
PACI	Public Authority for Civil Information

NOTATIONS

$\dot{Q}_d$	Heat gain by the ducts	(W_t)
$\dot{Q}_m$	Heat equivalent of the fan motor operation	(W_t)
$\dot{Q}_p$	Heat equivalent of the pump motor operation	(W_t)
$\dot{V}_a$	Volumetric flow rate of moist air	(m^3s^{-1})
$\dot{V}_w$	Volumetric flow rate of the chilled water	(m^3s^{-1})
A	Inside surface area of pipe	(m^2)
C	Present value capital cost	($)
C_{chil}	Nominal chiller capacity	(kW_t)
c_p	Specific heat of moist air	$(Jkg^{-1}{}^oK^{-1})$
CR_{char}	Capacity ratio when charging	-
CR_{dirt}	Capacity ratio when direct cooling	-
E	Present value energy costs	($)
e	Real escalation rate	-
E_f	Fan total efficiency	(%)
E_m	Motor efficiency	(%)
E_p	Pump total efficiency	(%)
F	Future value of one item amount	($)
F_{lm}	Motor load factor	-
F_{um}	Motor use factor	-
h_a	Air side convective heat transfer coefficient	$(Wm^{-2}{}^oK^{-1})$
H_{char}	Number of charging hours	(h)
H_{dirt}	Number of direct cooling hours	(h)
h_w	Water side convective heat transfer coefficient	$(Wm^{-2}{}^oK^{-1})$
i	Interest rate	-
i'	Effective interest rate	-
j	Inflation rate	-
k_i	Thermal conductivity of insulation	$(Wm^{-1}{}^oK^{-1})$
k_p	Thermal conductivity of pipe	$(Wm^{-1}{}^oK^{-1})$
L	Length of pipe	(m)
LCC	Total life cycle cost in present value of a given AC system	($)
OMR	Present value non-fuel operating, maintenance, and repair costs	($)
q	Heat transfer rate to water	(W_t)
r_1	Internal radius of pipe	(m)
r_2	Internal radius of insulation (for example, pipe outer radius)	(m)
r_3	External radius of insulation	(m)
Rep	Present value capital replacement costs	($)
T_a	Temperature of ambient air	(ºC)

TC_{char}	Total integrated cooling demand during charging time only	(kW_th)
T_i	Inlet chilled water temperature to the storage tank	(°C)
T_o	Outlet chilled water temperature from the storage tank	(°C)
T_w	Temperature of chilled water	(ºC)
U	Overall heat transfer coefficient	($Wm^{-2o}K^{-1}$)
UPV	Uniform present value factor	-
V	Volume of the tank	(m^3)
W	Present value water costs	($)
Δp_a	Fan total pressure	(Nm^{-2})
Δp_w	Pump total pressure	(Nm^{-2})
Δt_d,	Assumed temperature rise in the ducts	(°K)
ρ	Water density	(kgm^{-3})
ρ_a	Density of supply air or return air	(kgm^{-3})

Chapter 1

General Introduction and Thesis Overview

1.1 Background

Urban developments in Kuwait are associated with the extensive use of indoor AC systems. This is because of the long duration of the hot summer season, which lasts for more than seven months during which long spells of high temperature and high humidity conditions are encountered. Considering the above, a major part of the energy consumed in Kuwait is due to indoor AC. In its energy conservation programme code of practice guide, MEW (1999a) states that "Air conditioning installations absorb 60% to 70% of the consumed energy. As a result, the need for the construction of new power plants is important to satisfy this continually growing need.

In addition, AC in Kuwait utilises a high proportion of the country's generated electrical power. Al-Marafie et al. (1989) state that air-conditioning accounts for approximately 70% of the country's annual peak load and 45% of the yearly electricity consumption. With reference to power consumption data for the year 1999, it was found that nearly 75% of the annual load and 59% of the yearly electricity consumption is due to the use of AC systems in buildings.

The Ministry of Energy (MOE), the main supplier of electricity in the country, is deeply concerned about the excessive consumption of electricity due to AC. The ministry has enforced energy conservation measures since 1983 by applying a well-defined code of practice for new and retrofitted buildings. This code requires a minimum insulation of walls and roofs, restricts the glazing areas for a given type of glazing, specifies a minimum ventilation rate, and controls the performance rating of different types of AC systems. More importantly, by restricting the power requirement per unit area to remain below a specified value for a building and type of AC system, it helps the MOE to control the power supply to the buildings (MEW, 1999a).

By the end of 2001, the country achieved a total saving of over US$ 5.2 billion from reductions in the installed capacity of AC systems, and demand for power plants for the operation of AC systems and fuel required to generate the electricity needed to operate AC systems (Meerza, 2003). Although the 1983 energy conservation code is being applied to all new and renovated buildings, the maximum power and electricity demand between 1995 and 2001 grew at an annual rate of 6.9% and 6%, respectively (MEW, 2002).

1.2 Reducing the maximum power

Building pre-cooling and cool thermal storage systems are two basic approaches that can reduce the maximum power demand during day time. In the first approach, the building is pre-cooled at night or in the early morning during a period of low electricity demand, storing cooling in the building thermal mass and thereby reducing cooling loads during the maximum power demand in the day time. Savings are achieved by reducing the use of electrical energy and demand charges at day time. The potential for utilising the building thermal mass for load shifting and maximum power demand reduction has been demonstrated in a number of simulation, laboratory, and field studies (Braun, 1990; Snyder, 1990; Ruud et al. 1990; Keeney, 1997). However, the energy consumption of the AC in this approach was found to be high due to the lower temperature setting during the pre-cooling time.

Cool thermal storage is the second approach that can be implemented in the AC system to reduce the maximum power demand. Different cool storage technologies are available; they can be used to store cooling energy, and can be implemented within the AC system with different operating strategies (Dorgan, 1994; Dincer, 2002a).

Cool thermal storage has many benefits, which are discussed and reported in Dincer (2002b), and MacCracken (2003) including the economics, energy savings, design consideration and impact on the environment. The feasibility of incorporating cool thermal storage into conventional AC systems was studied in Hussain et al. (2004) and Al-Rabghi (2004) where cool thermal storage was considered as one of the available energy saving technologies.

Many countries, including the UK (Seaman, 2000) and US (Potter, 1995), have used cool thermal storage technology to shift the maximum electrical power requirements from periods of high demand to periods of lower demand. However, in the Gulf area, Saudi Arabia has only integrated cool thermal storage with the AC systems for the cooling of some buildings. A list of major projects using both chilled and ice thermal storage systems is reported in Hasnain (2000a). A preliminary study reported in Hasnain et al. (2000b) shows that thermal energy storage can reduce the peak cooling load demand by approximately 30%–40% and the peak electrical demand by approximately 10%–20%.

Although Kuwait was the first country in the Arabian Gulf to implement energy conservation measures in air conditioned buildings through well defined codes (Maheshwari, 2001a), a recent study shows that AC still consumes about 39% of the overall electrical energy generated from the power stations and about 62% at the maximum power demand on a peak summer day (Sebzali, 2006b). Therefore, more research and work needs to be done in order to reduce further the maximum power demand of buildings and possibly reduce the energy consumption by using cool thermal storage technology.

Recently, Sebzali and Rubini (2006a) studied the impact on the energy performance of air cooled chillers of using ice cool thermal storage with different operating strategies. Although Sebzali (2006a) found that ice cool thermal storage can reduce the electrical load during the peak periods of cooling load, the energy consumption increased by up to 8%, depending on the operating strategy, compared to a conventional AC system. The increased energy consumption of the chillers will additionally increase the operating cost of these types of system, because the cost of the electricity is directly proportional to the energy consumption by the cooling system, which in Kuwait does not vary between day time and night time. Moreover, Sebzali (2006b) examined the impact on the energy performance of air cooled chillers of using a chilled water thermal storage system, and showed that chilled water storage systems can reduce the maximum electrical demand of a chiller in an air cooled AC system by up to 100% and reduce the nominal chiller size by up to 33% depending upon the operating strategy adopted. This is achieved with only a 4% increase in energy consumption of the chiller for all chilled water storage operating strategies except for full storage where the energy consumption actually decreases by approximately 4%.

1.3 The aim of the thesis

This thesis aims to utilise cool thermal storage systems technologies and strategies within the AC system to reduce maximum power demand and energy consumption of an air-cooled AC system for a typical medium size office building in Kuwait, and to investigate their economic impact on the Kuwaiti government and users. This is through design and sizing of the components in the conventional and cool thermal storage AC systems, estimating and analysing the energy performance and the life cycle costs. In Kuwait, there is no cheap rate electricity tariff and there is no direct cash incentive offered by the Ministry of Energy for demand management measures. However, cool thermal storage may be attractive for both the user and the government if

a. The electrical power consumption during the maximum cooling load is reduced.
b. Chiller operation during the night time improves energy conservation and hence operating cost.
c. AC system and power connection costs are reduced.

A clinic building in Kuwait which represents a typical medium size office building in Kuwait (see Figure 1.1) was selected to assess the use of cool thermal storage on the electrical power and energy consumption of the air-cooled AC systems. The building is located within a hospital complex comprising two blocks, referred to as A and B, connected by a small corridor. Block A is a single story construction located at the rear part of the building. Block B has ground and first floors in addition to a tall reception area with a large glassed area including a skylight. This building is occupied from 7:00 hours to 14:00 hours for five days a week, and has a total floor area of 3180 m^2.

Figure 1.1 The clinic building.

1.4 Overview of the thesis

This thesis aims to study the energy and economic impact of using cool thermal storage technology on the air-cooled AC system of the clinic building. It discusses the design and selection of both conventional and cool thermal storage AC systems, determines the power and energy requirements, and calculates the life cycle cost for each AC system design and strategy. The calculation is based on the actual cost of electricity and connected power estimated by the Kuwaiti government and subsidised cost for the user. Therefore, this study aims to determine which of the several major options and strategies is best for the Kuwaiti and similar climates.

This thesis is divided into eight chapters and six appendices. Chapter 1 is the general introduction of the thesis, which describes the background, aims and thesis structure.

The aim of Chapter 2 is to aid comprehension of the aim and objectives of this work; thus, it is necessary to describe the characteristics of the electricity supply industry in Kuwait. This chapter reflects upon the reasons that inspired the objectives of this research work by focusing on the problem that arises regarding maximum power and energy consumption in Kuwait due to the use of AC systems.

An overview of the electricity supply industry in Kuwait is given. The energy consumption and the proportion of the maximum power used due to the operation of AC systems in Kuwait are estimated. Furthermore, the breakdown of the power demand of two existing buildings is discussed and compared with the recommended power rating of the AC systems of Kuwait.

Chapter 3 aims to introduce an overall presentation of cool thermal storage technologies, and their basic concept in brief, control and operating strategies that are associated with this technology. This chapter also reviews the advantages, disadvantages, types and relative merits of the different storage media. In addition, a selection of suitable types of the cool thermal storage technologies that are to be examined in this thesis is given. Finally, the chapter ends with several case studies for both ice and chilled water storage systems showing the advantages and design aspects of using such systems.

In Chapter 4, the hourly cooling load profile of the clinic building is generated using the ESP-r building energy simulation program. The input data for the simulations requirements is presented and the results of the simulation are analysed for all months in the year. Moreover, other profiles for different types of buildings are examined and a breakdown of the cooling load is presented.

Chapter 5 presents the development of the total cooling load profile. This chapter first discusses the methodology of sizing different components in conventional and cool thermal storage AC systems including chillers, storage capacity, AHUs and pumps and then calculates the heat gains by different auxiliary systems such as AHUs and pump motors, ducts and pipes that will be added to the building cooling load to determine the system cooling load profile for each examined AC system design.

Chapter 6 introduces the calculation of the power and energy consumptions of the AC systems for both conventional and cool thermal storage systems. In this chapter, the input power requirements for each AC system are calculated for the design day conditions and whole year. Furthermore, a results analysis for the proportion of the maximum power demand and energy consumption used by each component is presented for the AC systems and comparisons are made between the cool thermal storage systems with various operating strategies and the conventional system.

Chapter 7 presents the economical analysis of the cool thermal storage system. The capital, maintenance, and operating costs are estimated; then, the life cycle cost for each AC system is calculated. The results are compared, and the optimal storage technology and operating strategy is then specified.

Finally, Chapter 8 introduces the conclusions of this research; it summarises the key stages of the work and offers recommendations for future work.

Of the appendices, Appendix A calculates the psychrometric properties of the air distribution system. Appendix B gives the specifications of different components in the AC systems. Appendix C provides a detailed design and construction of the chilled water storage tanks. Appendix D discusses the design and distribution of the chilled water system. Appendix E gives the types, accessories, and fittings that are associated with different components in the AC system. Appendix F provides the tables of the chillers performance. Appendix G presents a breakdown of the capital costs of the AC systems. Appendix H lists the tables of the estimated life cycle costs of the AC systems. Finally, Appendix I. illustrates the control systems for the AC systems.

Chapter 2

The Supply of Electricity in Kuwait

2.1 Introduction

In Kuwait, electricity is becoming an increasingly essential element of modern life. This flexible energy is used for heating, cooling, appliances, TVs, computers, transportation, and a myriad of other uses. The demand for electricity in Kuwait is continuously increasing, growing at an average rate of 6.2% per year (Sebzali, 2006a) well above the world average of 2.7% (Khatib, 2005). As a consequence, in order to maintain a 20% reserve capacity, additional power plants must be constructed to satisfy this demand. It was reported (EIA, 2004) that Kuwait must invest around US$ 4 billion over the next ten years to finance an expansion programme for an additional 3.4 GW_e of electricity to cope with this predicted demand.

Kuwait is one of many countries in the world with a hot dry climate that relies heavily upon AC systems for the cooling of buildings. AC systems have become a necessity for the modern life style to provide adequate comfort and a healthy indoor climate. As will be shown later, the AC systems of buildings in Kuwait consume about 62% of the peak electrical power and 45% of the annual exported electricity

There are different methods by which the cooling effect is produced. The most common method to provide comfortable conditions in small, medium, and large buildings in Kuwait is by using package or split units, air-cooled and water-cooled chiller systems respectively. At the heart of all these systems are the compressors, which, in most cases require electric energy to operate. The electric compressors are considered to be the easiest and most preferred refrigeration method because the electric energy supply is widely available. For this reason, the demand for electricity is high in Kuwait especially in the summer months.

This chapter gives an overview of the annual and maximum electricity generation of Kuwait. The data discussed in this chapter was obtained from the latest statistical electrical energy year books published in years 2002, 2003, and 2004 by the MOE of Kuwait (MEW, 2002; MOE, 2003; MOE, 2004). The books contain all available information regarding the yearly and monthly electricity consumption of the individual power stations in the country. It also contains the future estimate of the installed capacity of the power stations, in depth electrical transmission networks and developments, and other information related to the fuel consumption of each power station.

Further analysis is conducted to estimate the proportion of the electrical power and energy consumption of the AC systems using the statistical electricity energy year books, as well as the hourly power generation collected from the control department of the Ministry for the year 2003.

This chapter also discusses the recommended power rating for both water and air-cooled AC systems for Kuwaiti design conditions, and estimates the proportion of the power consumption by each component such as chiller, pump, and AHU to maximum power demand in the AC system. Since cool thermal storage systems have a significant impact specifically on the power demand of the AC system at the peak demand time, the power demand for each component in two existing AC system buildings, the Kuwait Institute for Scientific Research (KISR) and the Public Authority for Civil Information (PACI) buildings, are examined and compared with the recommended power rating for Kuwait.

2.2 Patterns of electricity in Kuwait

In July 2003, the Water and Electricity Ministries of Kuwait were merged, along with the Oil Ministry, into a new Energy Ministry. The new Ministry is in charge of power and water provision, the management and operation of power plants, and general policy for the Kuwaiti energy sector.

Kuwait has five fossil fuelled power stations: Doha East, Doha West, al-Subiya, Shuaiba South, and az-Zour South. The percentage installed capacity for different power stations is shown in Figure 2.1 (MOE, 2004). The total installed electrical capacity is about 9.2 GW_e. The MOE expects that the installed electrical capacity will grow by about 5.9% to 10.2 GW_e by the year 2008.

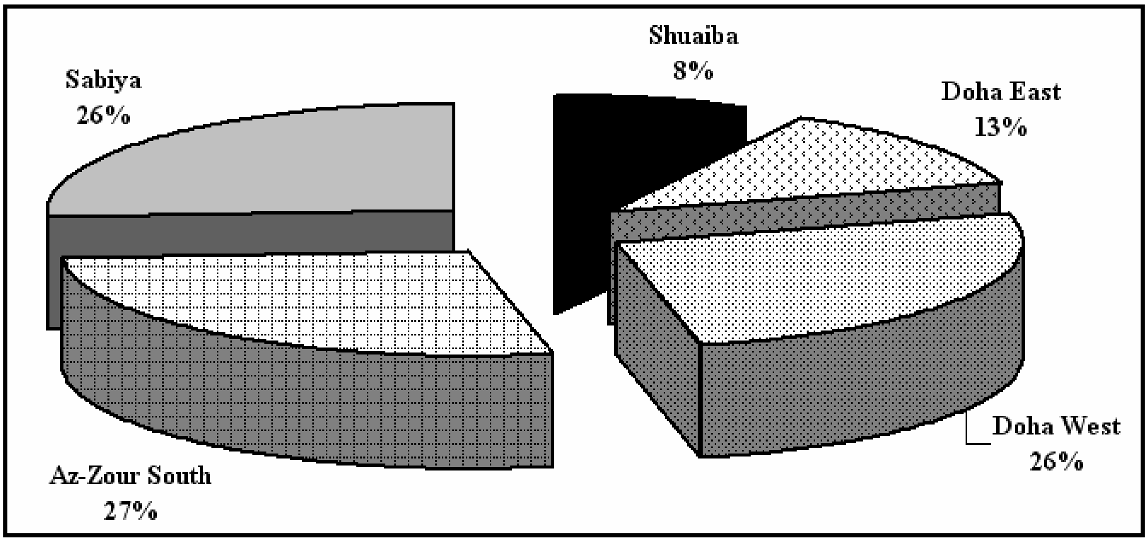

Figure 2.1 Percentage installed capacity of the power stations in Kuwait (MOE, 2004).

The generated fossil-fuelled energy consumption has increased in recent years mainly due to the heavy use of AC systems, reliance on desalination for water, and highly subsidised electricity and water prices. In 2003, the amount of natural gas consumed by power stations increased by 2.2% to 75.9 billion m^3 from 74.3 billion m^3 in 2002 (MOE, 2004). Furthermore, the consumption of liquid fuel gas also increased by 22.3% and that of heavy oils by 79.6% from 2002 to 2003. However, the consumption of crude oil decreased by 63.7%.

The proportion of electrical energy that is consumed by various sectors in Kuwait is shown in Figure 2.2 (MOE, 2004). The figure shows that 43% of electricity exported from the power plants in Kuwait is consumed for private purposes (e.g. resident houses). This is because the government and people are establishing more and more residential areas every year (Al-Hasan, 1997).

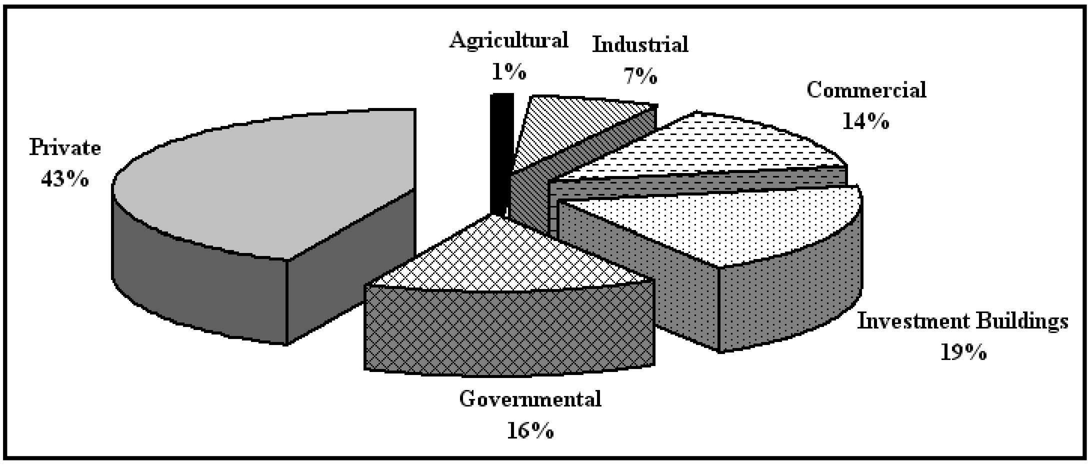

Figure 2.2 Proportion of electrical energy consumption by various sectors in Kuwait in 2003 (MOE, 2004).

Although the private sector has the maximum proportion of the electricity consumption, it decreased from 48% in 2002 to 43% in 2003. However, the proportion of electricity consumption for the investment buildings increased from 12% in 2002 to 19% in 2003 and that for the commercial sector increased from 10% in 2002 to 14% in 2003 (MOE, 2003; MOE, 2004). Furthermore, the proportion of the electrical energy consumption decreased from 2% to 1% for the agricultural sector, 11% to 7% for the industrial sector and from 17% to 16% for the government sector from 2002 to 2003.

Kuwait's per capita electricity consumption in 2003 was ranked number seven in the world (United Nations, 2005). In 2003, the estimated per capita consumption of electrical energy was 13.0 MW_eh per year and 35.6 kW_eh per day (MOE, 2004). Kuwait has some of the lowest power prices in the world and the government may reduce subsidies to try to cut its present electricity use. At present, all tariffs for domestic customers are straight line tariffs. Electricity costs about 0.069 US$ per kW_eh, while water costs about 0.74 US$ per m^3. The actual cost of electricity production was estimated to be 0.69 US$ per kW_eh, which means that electricity users in Kuwait pay only 10% of the actual cost of electricity.

The MOE also charges for the cable connection for newly constructed buildings. The cable connection requires a current capacity sufficient to feed power to a user up to the maximum power drawn from the utilities for the operation of the AC system. The buildings' owners pay a single payment for the cable connection only 169 US$ per kW_e; however, the actual cost was estimated by the Ministry to be around 1,356 US$ per kW_e. In fact, the cost of the cable connection is higher than this, because this cost represents only the cost of the power plants, transmission, and distribution, and excludes fuel and labour costs.

2.3 Annual electricity generation

The electrical energy generation increased rapidly by almost 70% from 22.8 TW_eh in 1994 to 38.7 TW_eh in 2003 as shown in Figure 2.3 (MOE, 2004). The figure also shows that the maximum annual electrical energy generation grew from 3.8 TW_eh in 1997 to about 4.2 TW_eh in 1998, which is about 9.5%. The minimum percentage growth is about 3.2% from 3.3 TW_eh in 1994 to 3.4 TW_eh in 1995. The average percentage increase of electrical energy for the ten years shown in the figure is approximately equal to 6.2%. This high average percentage yearly increase in electricity generation is due mainly to a heavy increase in the load of AC systems, which contributes to about 39.1% of the total yearly electrical energy generation in the country.

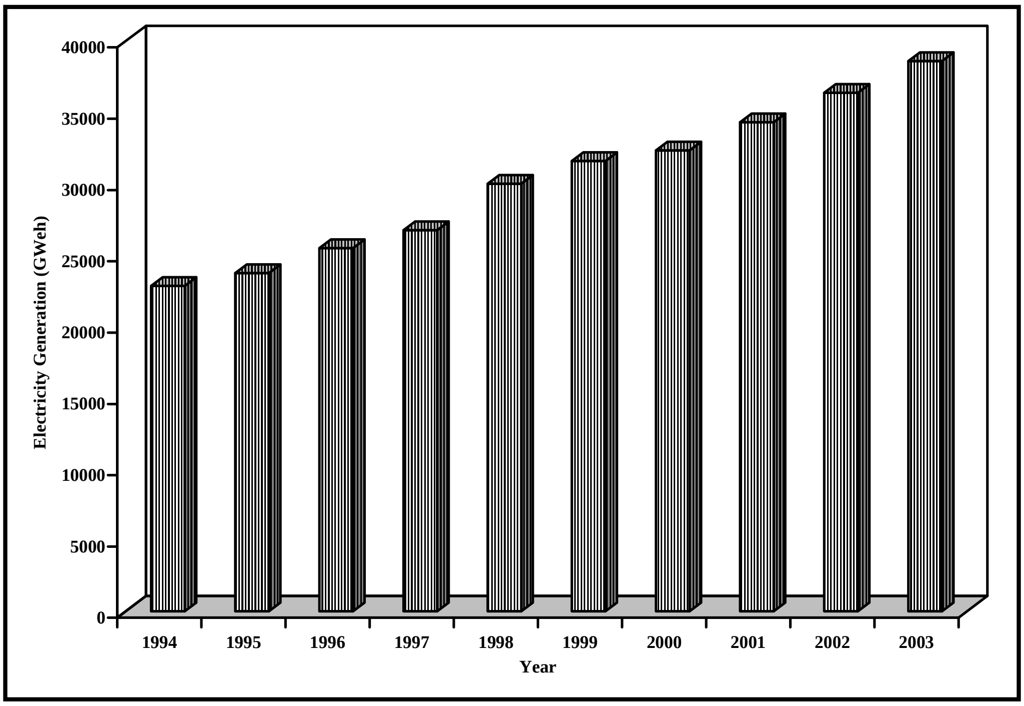

Figure 2.3 Annual electrical energy generation (MOE, 2004).

It very important to look at the maximum electricity generation of the country since the size of any new power stations to be built mainly depends on the expected maximum power withdrawal from the power stations. The peak or maximum electricity generation in MW_e in 1994 to 2003 is shown in Figure 2.4. The figure shows that the maximum power increases year after year; this is due to the significant increase in the number of new buildings associated with a heavy demand by the AC system, which always occurs at the peak summer months such as June, July, August, and September when the dry bulb temperature is very high and sometimes reaches above 50° C.

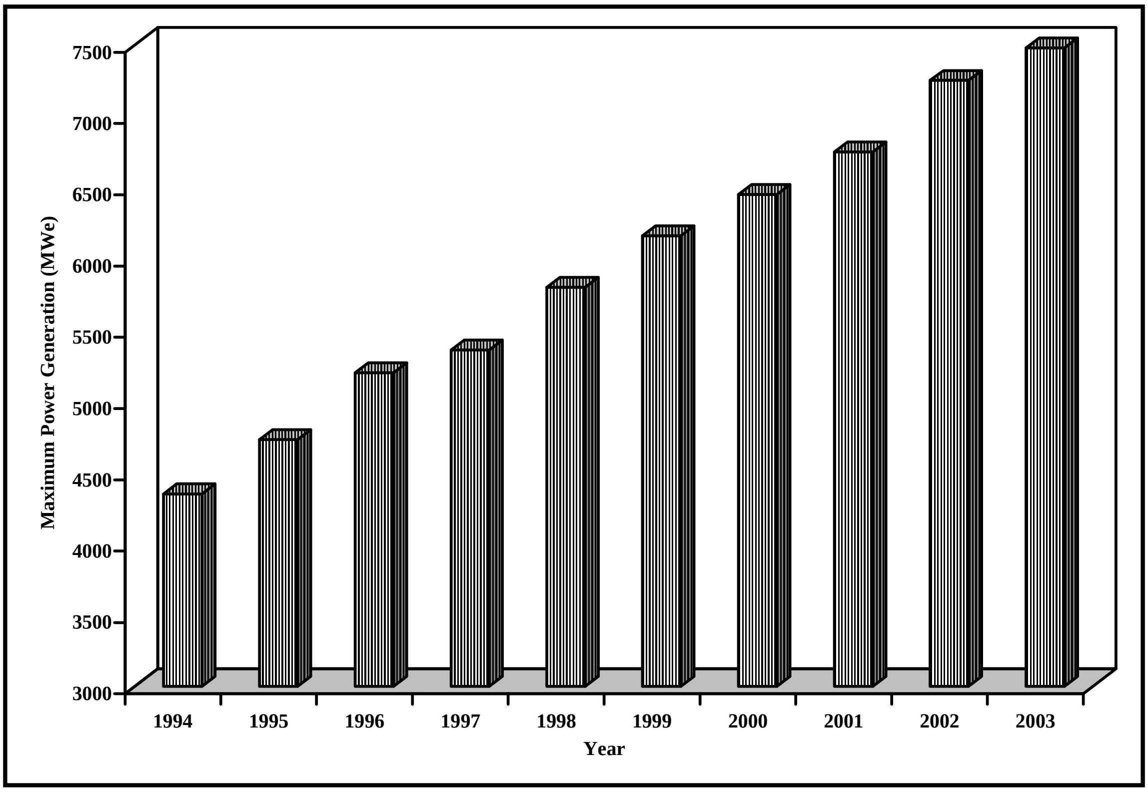

Figure 2.4 Annual maximum electricity generation in Kuwait (MOE, 2004).

As shown in Figure 2.4, the maximum electricity generation in 2003 was about 7.5 GW_e an increase of about 72% compared to the maximum electricity generation of 4.4 GW_e recorded in 1994. This means the capacity of the existing power stations in 2003 was increased by approximately 70% compared to the existing power stations in 1994.

It is equally important to look at the percentage of the utilisation factor, which is defined as the ratio of peak demand to the total installed capacity multiplied by 100 for the existing power stations. A plot of the utilisation factor for 1994 to 2003 is shown in Figure 2.5 (MOE, 2004). Figure 2.5 is an important figure since it gives the country information about when new power plants must be built. The higher percentage of the peak electrical demand to the total installed capacity of the five stations means the country must invest to build new power plants to cope with the increase in the electrical peak demand.

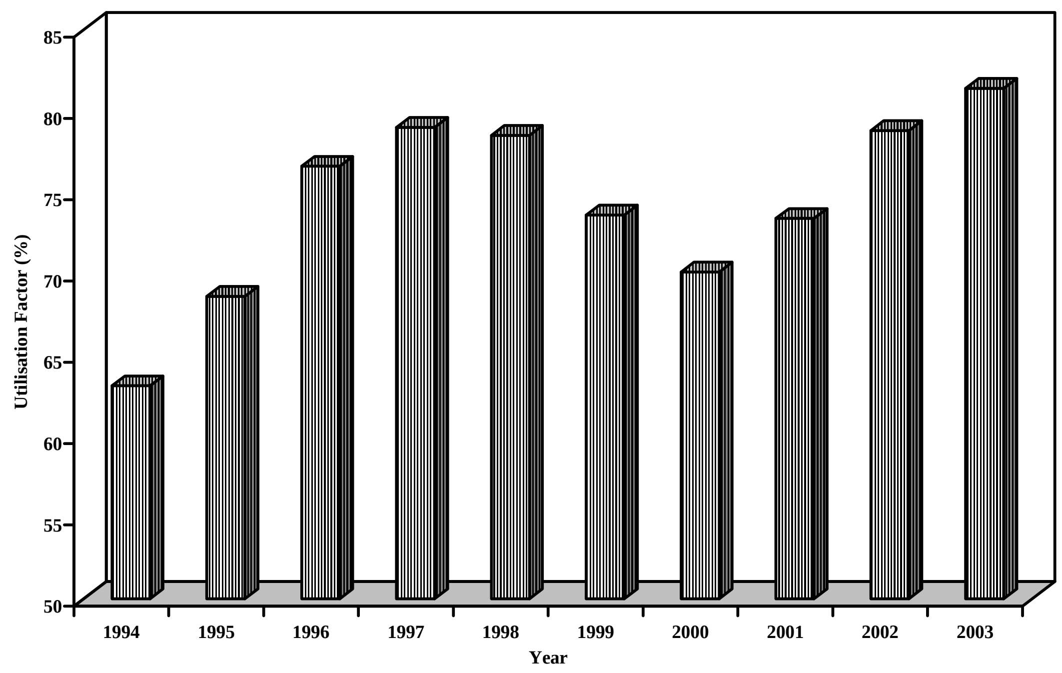

Figure 2.5 Utilisation factors of the existing power stations in Kuwait (MOE, 2004).

In Figure 2.5, it can be seen that the percentage of peak load to the installed capacity continuously increased from 1994 to 1997, and reached about 79% in 1997, and then slightly decreased in 1998 to 77.4%, and continued to decrease to about 70.2% in 2000. This is because in April 1998, Subiya power plant was partially commissioned at a cost of around US$ 2.2 billion, and in 2000, the power plant operated at full load with 2.4 GW_e total capacities (MOE, 2004), thus increasing the total installed power generation capacity of the country and therefore reducing the utilisation factor.

The figure also shows that from year 2000 the percentage started gradually to increase up to 81.4% in 2003, which means more power stations must be built or expanded in the next one or two years to cope with the high electricity demand. Although there is still no statistical data available for 2004 and 2005, it is expected that the utilisation factor shown in Figure 2.5 will have decreased in 2004 and 2005 because a new gas turbine power plant in the az-Zour south power station, which consists of 8 simple gas turbine electrical generating units of approximately a total capacity of 1.0 GW_e was partially commissioned in the summer of 2004, and fully commissioned by the summer of 2005, thus increasing the total installed capacity.

2.4 Monthly electricity generation

The electrical energy generation by the power stations in GW_eh for each month in 2003 is shown in Figure 2.6. This shows a sharp increase in the electrical energy generation during the summer months, because of the high demand of AC systems, and requirement of heat input for the production of distilled water resulted in a huge load on the power stations. The highest electrical energy generated was in August with 4.8 TW_eh, and lowest was in February with 1.8 TW_eh.

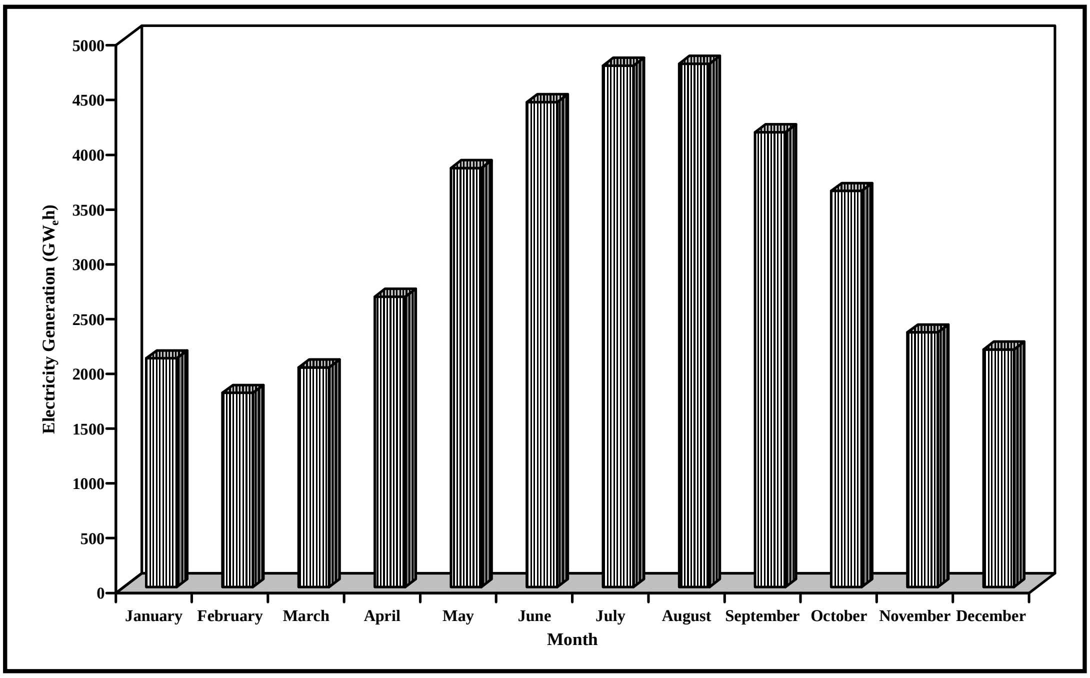

Figure 2.6 Monthly average electrical energy generations in 2003 (MOE, 2004).

Analysis of the electrical energy generation data obtained from the MOE for other years, for example, 1997 to 2002 shows that August always has the highest electricity generation because of the high average dry bulb temperature associated with this month, which ranges between 32 to 46° C, as well as the humidity compared to other months. Furthermore, February always has the lowest electrical energy since the weather conditions are mild for most of the days and there is only light use of the heating systems.

For the same year, the maximum electrical load in MW_e for each month was analysed and the results are shown in Figure 2.7 (MOE, 2004). The maximum electrical load occurred in July with 7.5 GW_e and a minimum of 2.1 GW_e in February. The maximum load in July represents about 81.4% of the total installed capacity of the country of that year.

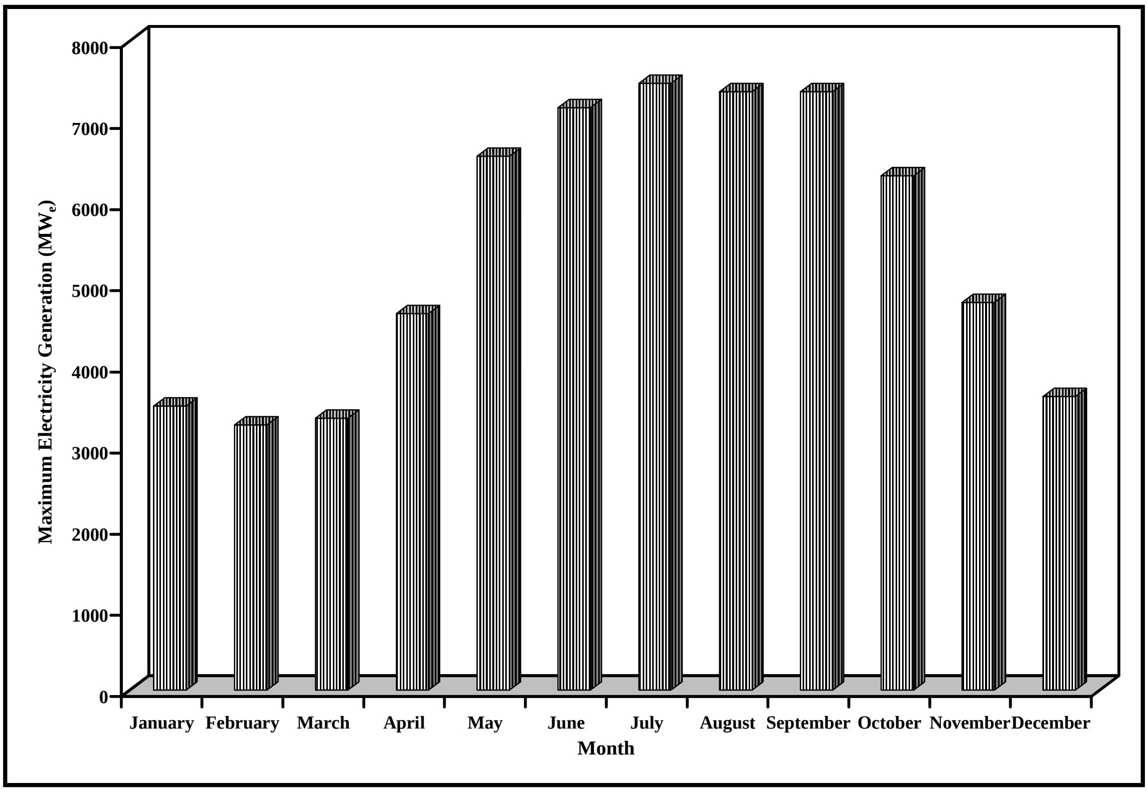

Figure 2.7 Monthly maximum electricity generations in 2003 (MOE, 2004).

In the winter season (December, January, and February), when the dry bulb temperature ranges between 10° C to 25° C for most days, the AC systems for all residential buildings that use package systems are normally switched off. However, in some large commercial, institutional, and governmental buildings, the AC systems are in operation throughout the year, and because of the moderate dry bulb temperature during these months, most of the AC systems are partially operating or they are operating at part load, which results in a low demand for electricity during these particular months.

2.5 Peak day electricity generation

It is essential to analyse the hourly electricity generation profile for a day at the peak of summer to know the time of maximum and minimum power demand for electricity. This also will help to select the charging and discharging times of the cool thermal storage, as will be discussed in Chapter 3.

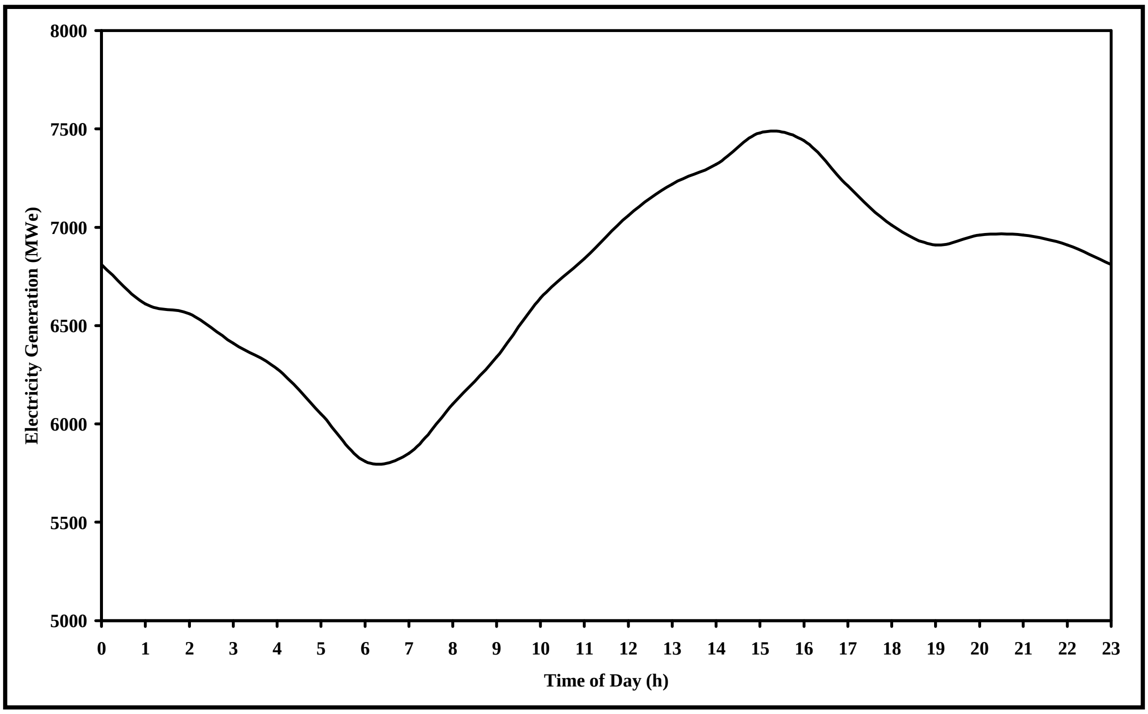

Figure 2.8 Hourly electricity generation at peak summer day in 2003 (MOE, 2004).

The hourly generation of electricity in MW_e for the peak day in 2003 is shown in Figure 2.8 (MOE, 2004). The high electrical demand of the day occurred between 12:00 hours and 16 hour and reached a maximum at 14:00 hours when the power stations generated the electricity at about 7.5 GW_e; the recorded dry bulb temperature at this time was 49° C and the relative humidity was 27%. The lowest demand was 5.8 GW_e occurring at 5:00 hours. The power demand rose gradually from the minimum at 5:00 hours until 15:00 hours due to the rise in the dry bulb temperature, and the corresponding increase in power demand of the AC systems of the buildings. After 15:00 hours, the power fell until 18:00 hours and then increased slightly at 19:00 hours at which time the lighting in buildings and streets are usually switched on during the summer. After 19:00 hours, the generated electricity slightly decreased until 23:00 hours; after that, it gradually decreased to the minimum at 5:00 hours.

2.6 Electricity consumption of the AC systems

It is difficult to determine exactly the energy consumption of AC systems in buildings throughout Kuwait. However, a reasonable estimate can be obtained using the statistical published electricity data (MEW, 2002; MOE, 2003; MOE, 2004) and actual measurements of the hourly electricity generation throughout the year provided by the control department in the MOE. Using these data, an approximation of the annual electricity generation in GW_eh, and the maximum power demand in MW_e at the peak time of the day can be estimated.

2.6.1 Annual AC system electricity consumption

To estimate the annual electrical energy required for the operation of AC systems, the monthly exported electrical energy data from power stations in Kuwait for 2001, 2002 and 2003 (MEW, 2002; MOE, 2003; MOE, 2004) was analysed. Because a proportion of the total electricity generated is used to drive the auxiliary units within the power stations and to produce distilled water, only the net exported electrical energy from the power stations was employed to estimate the annual energy consumption by AC systems. The exported electrical energy is defined as the difference between the electrical energy generated and the electrical energy consumed by the auxiliary units and the production of distilled water.

Using the exported electrical energy data, an estimate of the electrical energy required by AC systems was obtained based on the assumption that the exported electrical energy in the month of February is used neither for heating nor for cooling. This was based on the observation that February has the lowest electrical energy use compared to other months in the year.

Taking the difference between exported electrical energy in February and that of other months and assuming that energy in January and December is utilised for heating only, then the summation of the load from March to November may be assumed to represent the energy use by AC systems in buildings. Applying this simple calculation to the data for 2001, 2002, and 2003, the electrical energy consumption by AC systems was determined to be 13.0, 14.2, and 15.0 TW_eh respectively, as shown in Figure 2.9 (MEW, 2002; MOE, 2003; MOE, 2004), increasing by 14% in only two years. The average annual percentage of the electrical energy consumed by AC systems over three years, as shown in Figure 2.9, is 45% of the total exported and 39% of the total generated energy from all power stations in Kuwait.

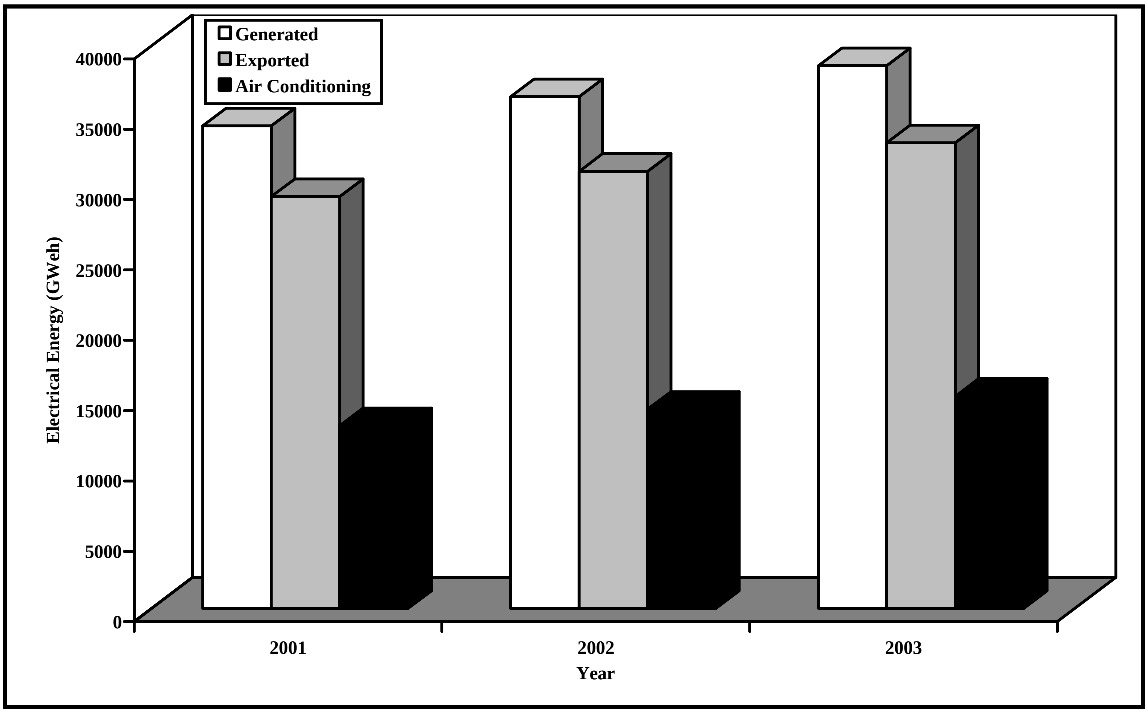

Figure 2.9 Annual electrical energy generated, exported and consumed by AC systems in Kuwait (MEW, 2002; MOE, 2003; MOE, 2004).

2.6.2 Monthly electricity consumption of the AC systems

From Figure 2.10, the proportion of electrical energy used for the operation of AC systems in the year 2003 was determined to be 46% of the exported and 39% of the energy generated. If it is assumed that the cooling and heating systems of buildings are not used in February, then the exported electrical energy in February represents the demand from other electrical devices. This is constant throughout the year and represents about 53% of the total exported electricity. This would appear to be a significant proportion, but in fact is a reasonable estimation because this represents not only the power consumption from different appliances in buildings and street lighting, but also the operation of major industries such as factories, refineries, and large petrochemical companies.

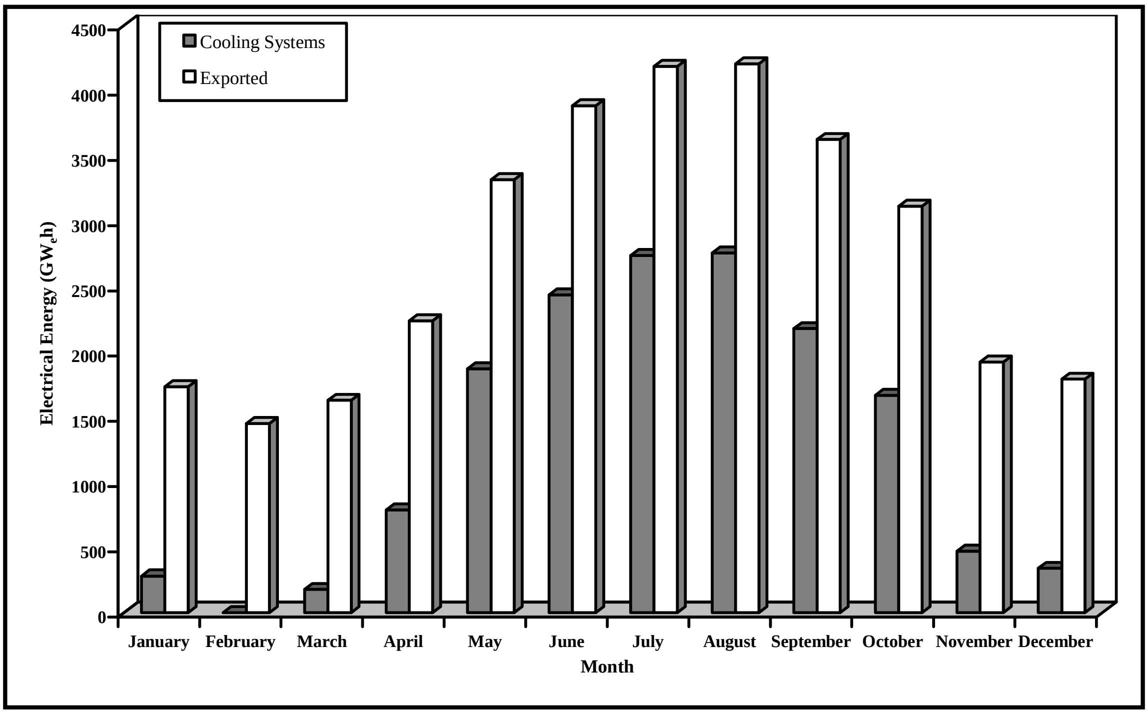

Figure 2.10 Monthly exported electrical energy and AC systems consumption in 2003 (MOE, 2004).

2.6.3 *Electricity demand of the AC systems at peak day*

An estimation of the electrical power demand by AC systems in buildings at the peak time for 2003 cannot be performed with the available published data (MOE, 2004). This is because the time at which the minimum and maximum of generated and exported electricity demand occurred was not reported. Furthermore, only an hourly load profile at the peak load day was provided and not an hourly profile at the minimum load day or even maximum days for other months in the year. However, the hourly electrical load for every power station in Kuwait was available. This data was obtained for 2003 to provide an estimate of the electrical load of the cooling systems in Kuwait.

To provide an accurate estimation, the original intention was to follow the methodology that was applied by a previous study (Al-Hasan, 1997). In this study, the maximum electrical load in March was taken as the load when electricity was not used for heating or cooling. It was also assumed that the difference between the maximum loads in the peak summer periods occurring in either July or August, and the maximum loads in the month of March, represented the energy consumed by AC systems. However, after careful analysis of the hourly power demand for the months of February, March, and August, it was decided this assumption might not be valid. Firstly, the peak power demand in March or February did not occur at the same time as in August. Based on the hourly average electricity demand values, the peak demand in February and March occurred at about 18 hours and that for August occurred at 14:00 hours as shown in Figure 2.11. Secondly, in March, AHUs in AC systems are frequently used, especially during the afternoon even though the average dry bulb temperature reaches only 22.3° C. For these reasons, February was selected instead of March as the base month.

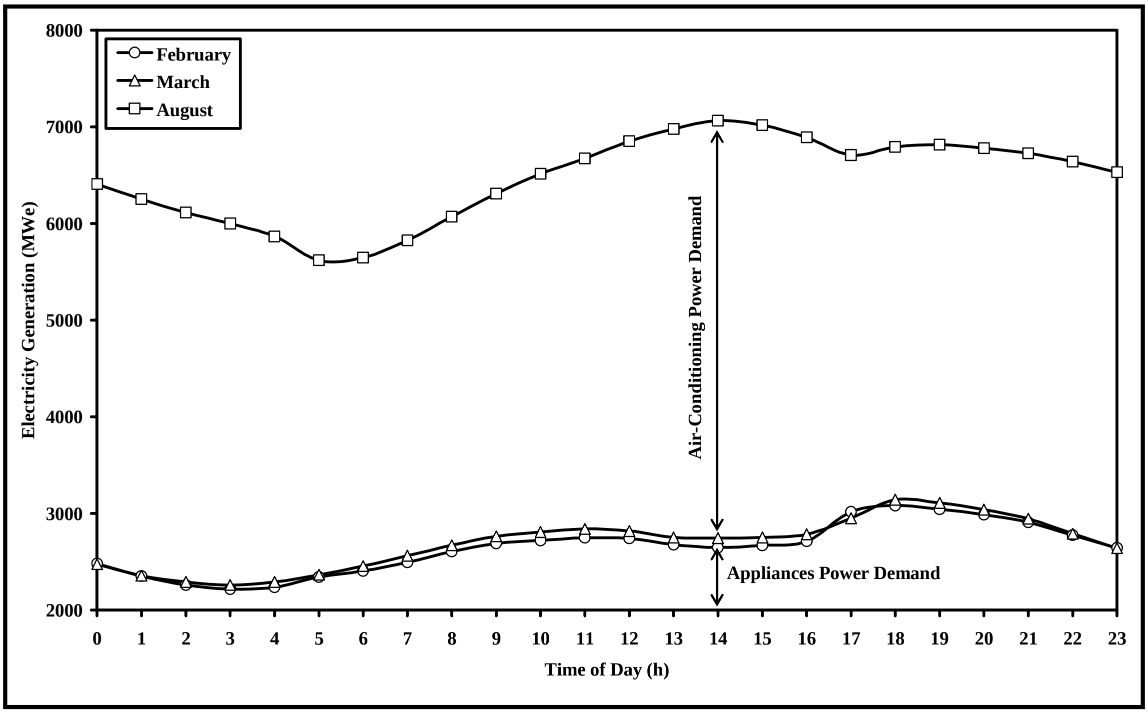

Figure 2.11 Actual average hourly electricity generation for February, March and August in 2003.

Figure 2.11 shows that the difference in the electricity demands between August and February at the peak time of 14:00 hours, implies that the power requirement of AC systems is about 4.4 GW_e. This represents about 62% of the average peak load in August. The demand for electricity in February at 14:00 hours represents the power consumed by different appliances, auxiliary units within the power stations and the production of distilled water, which is estimated to be about 2.7 GW_e. The above analysis was conducted based on the average of the actual hourly generated electricity data for each month; however, the maximum generated electricity was about 7.5 GW_e which occurred in July (MOE, 2004).

2.7 Power demand of AC systems in Kuwait

Air and water-cooled water chillers are commonly used for AC of medium and large sized buildings in Kuwait. Air-cooled chillers are generally favoured for medium size systems and water-cooled for large systems. In these systems, the chillers cool the water, and the cooled water is pumped through the piping system to the various air handling and fan coil units in the building. The fan in the air handling and coil units circulates the air in the ducting system and within the conditioned space. The return air from the room is then cooled and dehumidified by the cooling coils and returns again to the room.

Basically, in this process, the circulated chilled water in the cooling coil picks up the heats that is gained by the room, supply and return ducts, and fan motors, and is returned to the chiller for cooling in the evaporator. The refrigerant in the evaporator absorbs the heat of the return chilled in the chiller as it evaporates at a low temperature and pressure, and gives up this heat to the atmosphere as it condensates at a higher temperature and pressure.

2.7.1 Recommended power rating of AC system in Kuwait

The recommended power rating in kilowatt electricity per kilowatt cooling for air-cooled and water-cooled AC systems at the design conditions for Kuwait is shown in Table 2.1 (Maheshwari et al. 2003). The table was generated based on a market survey of the latest AC equipment manufactured or marketed in Kuwait, an experimental investigation conducted on a number of systems of different capacities and types and the reality of weather conditions. The survey was based on representative climatic conditions for Kuwait with a dry bulb temperature of 48° C and a wet bulb temperature of 32° C for air- and water-cooled systems respectively, a chilled water flow rate of 40 cm^3s^{-1} and inlet and outlet chilled water temperatures of 12.8° C and 6.7° C respectively (Maheshwari et al. 2003).

Cooling System		Power Rating (kW_e per kW_t)					
Type	Size (kW_t)	Chiller	Cooling Tower Fan Motor	Condenser Pump Motor	Chilled Water Pump Motor	Air Handling Units Motor	Total
Air-cooled	< 352	0.455	0.000	0.000	0.020	0.108	0.583
	352 - 879	0.455	0.000	0.000	0.020	0.108	0.583
	> 879	0.455	0.000	0.000	0.020	0.108	0.583
Water-cooled	< 879	0.270	0.011	0.017	0.020	0.108	0.426
	879 - 1759	0.213	0.011	0.017	0.020	0.108	0.370
	> 1759	0.199	0.011	0.017	0.020	0.108	0.355

Table 2.1 Recommended power rating of the AC systems in Kuwait (Maheshwari et al. 2003).

In an air-cooled system, the power demand of the chiller is a strong function of building load and ambient dry bulb temperature. However, the power demand of the AHU and chilled water pumps (if constant speed motors are used in the system) are generally constant during the whole summer period irrespective of the building cooling load and weather conditions.

In water-cooled systems the main components in the AC system that consume electricity are similar to those of the air-cooled systems but with the addition of the condenser water pumps and motors of the cooling tower fans. Table 2.1 shows that for larger water-cooled systems the power requirement of the chiller at peak load is lower and for other components is higher. The table also shows that the share of the power demand of the AHUs ranges between 25% and 30%, which is greater than for air-cooled systems. Furthermore, the average power rating of water-cooled systems is approximately 34% less than the air-cooled systems.

Based on the study conducted by Maheshwari et al. (2003), the peak power requirements and seasonal energy requirements of water-cooled systems are less than those of air-cooled systems by 17% to 47% and 23% to 44% respectively depending on the location, type, and building cooling load. Although water-cooled systems have a lower peak power requirement and lower energy consumption, they have higher life cycle costs compared to air-cooled systems because of high maintenance costs and the added cost of desalinated water (Maheshwari et al. 2003). It is for this reason that the use of cool thermal storage is more attractive with air-cooled systems than with water-cooled systems, since this will further reduce the peak power demand, whereby approximately 78% of the total power of the cooling system is due to the chiller.

2.7.2 The peak power demand of existing buildings in Kuwait

It is important to determine the actual electrical power requirement for different components in an AC system for an existing building at the peak day time period. Since this study considers only the use of cool thermal storage incorporated in a conventional air-cooled AC system, the following sections illustrate the electrical power demand of each component for two existing buildings. The individual contributions to the total electrical demand at the peak time period are highlighted.

2.7.2.1 Kuwait Institute for Scientific Research (KISR) main building

The AC system of the Kuwait Institute for Scientific Research (KISR) main building was commissioned in 1984 and consists of 10 air-cooled chillers, 4 chilled water pumps, 23 supply and 15 return air fans together with 67 fan coil units, with an estimated total connection load of 4.9 kW_e (Maheshwari et al. 2001b). The connected load for each component of the cooling system in the KISR main building is given in Table 2.2 (Maheshwari et al. 2001b).

Component Name	Quantity	Connected Load Per Unit (kW_e)	Total Load (kW_e)
Chiller	10	418	4180
Chilled Water Pump	4	75	300
Air Handling Unit (Supply Fan Motor)	23	0.55 to 45	258
Air Handling Unit (Return Fan Motor)	25	0.25 to 22	114
Fan Coil Units Motor	67	0.18	12

Table 2.2 Connected powers for different components in KISR main building (Maheshwari et al. 2001b)

The system is designed to cool the building during the peak summer period by the operation of 8 chillers, 3 chilled water pumps and all air handling and fan coil units. However, during the summer season 2000, only 6 chillers were in operation; 3 pumps and all air handling and fan coil units were continuously running. It is estimated that the total cooling production of the chillers is about 3.2 MW_t. The measured annual energy consumption of the AC system including heating is 9.8 GW_eh per year, which represents 69% of the total energy consumption of the KISR main building (Maheshwari et al. 2001b).

Based on the measurement of the voltage, current, and power factor for the motors of the chillers, chilled water pumps, air handling units and fan coil units, the actual power at the peak demand period was estimated to be 2.1 MW_e. The individual contribution for each component of the AC system in the building is shown in Figure 2.12 (Maheshwari et al. 2001b).

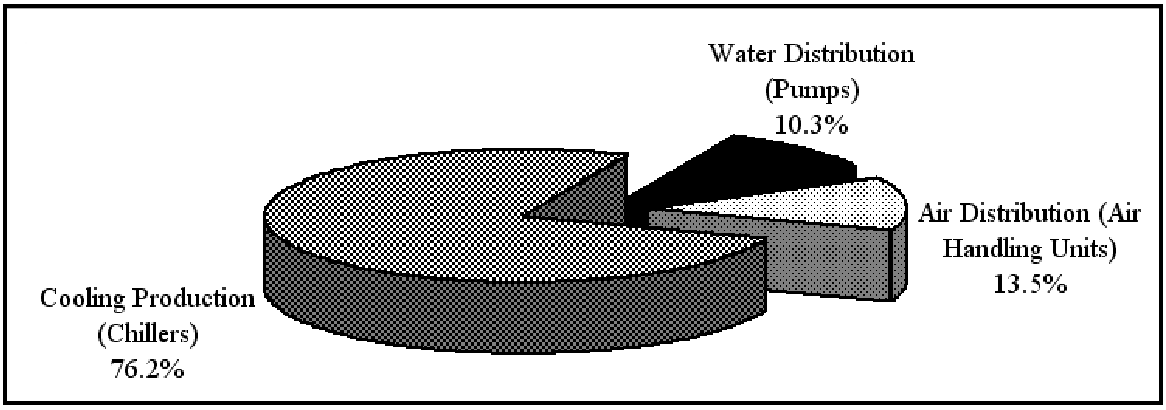

Figure 2.12 Actual percentage electrical power consumption at peak load in KISR main building (Maheshwari et al. 2001b).

The chillers consume the maximum power in the cooling system, about 76% of the total power at the peak period for cooling load. The air handling units and chilled water pumps represent only about 13% and 10% respectively. Compared to the recommended power rating discussed above, it can be observed that the power demand of the pumps is much higher than the recommended value of 3.4% given in Table 2.1. It can be seen from these actual measurements that the maximum power requirements for the chillers are during the peak cooling load period.

2.7.2.2 <u>Public Authority for Civil Information (PACI) building</u>

PACI comprises two buildings, the administration and the registration buildings. The administration building has a daily occupancy between 7:00 hours and 14:00 hours, while the registration building has a daily occupancy between 7:00 hours and 21:00 hours. The cooling load of the buildings is met at the peak summer period by the operation of 5 air-cooled chillers, 5 primary chilled water pumps, 19 AHUs, and 51 fan coil units, including 20 exhaust fans. The total estimated power demand of the AC system was 1.7 MW_e. A breakdown of the power connection for each component in the cooling system of PACI is shown in Table 2.3 (Maheshwari, 2005).

<table>
<tr><th rowspan="3"></th><th>Component Name</th><th>Quantity</th><th>Connected Load (kW_e per Units)</th><th>Total Connected Load (kW_e)</th></tr>
<tr><td>Chillers</td><td>5</td><td>261.9</td><td>1310</td></tr>
<tr><td>Primary Pumps</td><td>5</td><td>14.9</td><td>75</td></tr>
<tr><td rowspan="2">Air Handling Units</td><td>Supply</td><td>8</td><td>0.5 to 22.0</td><td>154</td></tr>
<tr><td>Return</td><td>8</td><td>6.3 to 9.0</td><td>49</td></tr>
<tr><td rowspan="3">Air Handling Units (Computer Room)</td><td>Fan</td><td>3</td><td>5.6</td><td rowspan="3">72</td></tr>
<tr><td>Reheat Coil</td><td>3</td><td>15</td></tr>
<tr><td>Humidifier</td><td>3</td><td>3.4</td></tr>
<tr><td rowspan="3"></td><td>Fan Coil Units</td><td>51</td><td>0.08 to 0.96</td><td>22</td></tr>
<tr><td>Exhaust Fans</td><td>20</td><td>0.01 to 1</td><td>4</td></tr>
<tr><td>Total</td><td></td><td></td><td>1686</td></tr>
</table>

Table 2.3 Power connections of PACI buildings (Maheshwari, 2005).

The individual contributions of the connected power of the components in the AC system such as chillers, chilled water pumps, air distribution system, and exhaust fans are shown in Figure 2.13. Again, as in the case of KISR main building, the greatest power demand is made by the cooling production (i.e. chillers), as it represents about 78% during the peak demand period, and is similar to the recommended power demand discussed above. The air distribution consumes about 17.6% of the peak power demand and is slightly lower than the recommended power rating given above. The contribution of the water distribution system is 4.4%.

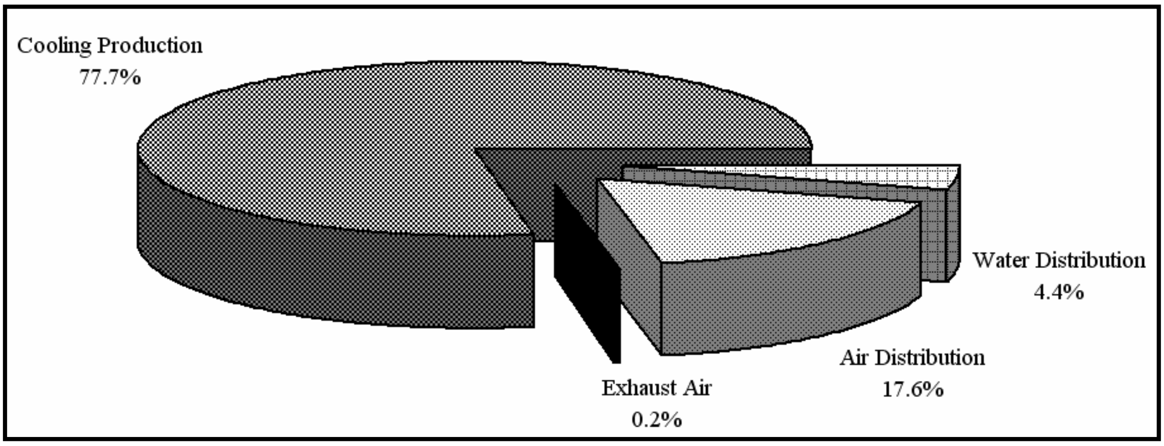

Figure 2.13 Actual percentage electrical power consumption at peak load in PACI (Maheshwari, 2005).

2.8 Conclusions

It has been shown in this chapter that the demand for electricity in Kuwait is growing annually at an average of 6.2% because of the continual expansion in building construction, urban development, high cost of subsidised energy cost and the reliance on AC systems for the cooling of buildings. It has also been shown that the maximum electricity generation has grown by about three quarters in a period of only ten years, so a need for constructing new power generation plants almost every few years is a big challenge for the Kuwaiti government.

Constructions of modern offices and large commercial buildings in Kuwait are always associated with the installation of large AC systems. AC systems are the major energy consumers in buildings. It has been estimated the AC systems in Kuwait annually consume about 39% of the total electrical energy generated and 45% of the exported electrical energy. It has also been estimated that approximately 63% of the electricity generation during the peak demand period is consumed only by the AC systems of buildings.

Based on the actual power measurement and connected power for some of the existing buildings in Kuwait it has been found that the motors of the compressors in the chillers have the maximum power withdrawal compared with other AC components. In air-cooled systems, the chillers consume about 76% of the electricity at the peak demand. Therefore, incorporating cool thermal storage specifically in the design of air-cooled systems for a medium building size will have a great advantage, since cool the thermal storage will have a greater effect on reducing the power demand of the chillers compared to other components especially at the peak demand period.

Chapter 3

Cool Thermal Storage Systems Technology

3.1 Introduction

Cool thermal storage systems can be used to reduce significantly the peak power demand for the AC systems in buildings by allowing energy intensive electrically driven chillers to operate mostly during night time when the electricity rates and demand are lower. In addition, some configurations and designs may result in lower capital costs and/or lower operating costs. In recent years, a number of design options have made cool thermal storage more energy efficient than conventional systems. Depending on the system configuration, the chiller may be smaller than would be required for direct cooling, leading to smaller auxiliaries such as the cooling tower, the condenser and the chilled water pumps. Pumping energy can be reduced by increasing the chilled water temperature range, and fan energy can be cut with colder air distribution. Al-Rabghi (2004) argued that thermal energy storage could be one of several technological methods for lowering the energy consumption of buildings if it were incorporated within their AC systems.

This chapter describes the basic types and features of cool thermal energy storage systems technologies such as ice, chilled water, and eutectic salt, and illustrates the basic strategies of using cool storage systems to reduce on peak electricity use such as full storage, partial storage, storage and chiller priority strategies. An appropriate selection of storage systems types is discussed for Kuwait. Furthermore, detailed discussions of the energy performances of pumps, storage tanks and chillers as well as system configuration and design are given. Finally, the chapter ends with several case studies for both ice and chilled water storage systems showing the advantages and design aspects of using such systems.

3.2 The cool thermal storage systems

Air conditioning systems incorporating cool thermal storage can help the electrical utilities reduce the load during the on peak period and increase the load during the off peak period. This shifting of the load improves the utilisation of base load generating equipment. Both electric utilities and customers can benefit from such systems. Electric utilities benefit from a reduction in peak demand, and customers benefit from lower electricity bills, depending on the electrical rate structure, as the storage systems allow customers to take advantage of lower off peak rates for electric energy and thus reduce peak demand billing charges.

In a conventional AC system, the electrical power demand is proportional to the building cooling load, which typically reaches a maximum during the afternoon period, thus contributing to peak power demand on the electric utility. The advantages of a cool thermal storage system over a conventional AC system are summarised below (Hasnain, 1998):

a. Compared with a conventional AC system, the refrigeration capacity can be substantially reduced, as it no longer has to cope with the peak building load.

b. The chiller plant operates at 100% of its rated capacity, throughout its period of operation in the cool storage system. Therefore, the plant works at its optimum efficiency, unlike conventional systems, which operate on part load operation for most of their working life.

c. Cool storage systems shift the refrigeration electricity load to the night time and the ambient temperatures at night time are substantially lower than those in the day time. Consequently, there is an improvement in chiller efficiency and a constant generating load can be maintained, ensuring efficient use of the plant compared with day time operation.

d. The larger air temperature difference over the AHUs allows a reduction in circulated air volume, results in smaller AHUs, smaller piping air handlers and duct work, less electrical equipment and wiring, and also reduces the size and cost of the duct work.

e. The installation of cool thermal storage systems in buildings significantly reduces chiller plant capacities and, consequently, the size of refrigerant gas charges, thereby reducing the emission of harmful CFCs into the atmosphere, which should help reduce both the depletion of the ozone layer and the formation of the greenhouse effect.

Thousands of cool storage systems have been installed in the United States. A survey conducted for ASHRAE resulted in an estimated population of 1500–2000 systems in the early 1990s (Potter, 1995). Applications cover a wide range of facility types, but most commonly are offices, schools, department stores and hospitals.

Between 80% and 85% of the systems installed use one of the several kinds of ice storage. Another 10%–15% use chilled water storage, with eutectic salt systems representing about 5% of the systems in the survey conducted by Potter (1995). Based on that survey, the actual demand reduction and cost saving for ice thermal storage were calculated as being 100.9% and 117.4% respectively, with the greatest benefits being achieved with chilled water and eutectic salt systems. It has also been found that most of the ice storage systems were installed for relatively small buildings with typical energy storage requirements ranging from 100.8 MW_th to 0.4 MW_th, and that these had the highest cost in US$ per kW_th compared to other storage technologies.

3.3 Cool thermal storage types

There are many different types of cool thermal storage systems. They represent various combinations of storage media, a charging mechanism, and a discharging mechanism. The available basic media options are water, ice, and eutectic salts. Ice storage systems can be further broken down into ice harvesting, ice-on-coil, ice slurry, and encapsulated ice options (Dorgan, 1994).

3.3.1 Ice Thermal storage systems

In ice thermal storage, the latent heat of fusion for water of 334.4 $kWhkg^{-1}$ is utilised to store cooling energy. The storage volume is generally in the range of 0.019 to 0.027 $m^3kW_t^{-1}h^{-1}$ depending on the specific ice storage technology (i.e. ice harvesting, external melt ice-on-coil, internal melt ice-on-coil and so on). Since an ice storage system stores cooling energy in ice at 0°C, the freezing point of water, the chiller must be capable of producing charging temperatures of -9.0° C to -3.0° C, below the normal operating range of conventional chillers. The heat transfer fluid in the ice thermal storage systems may be the refrigerant itself or a secondary coolant such as 25% or 30% ethylene glycol mixed with water.

A chilled water temperature of 1.0° C can be produced by using ice storage systems. Through a proper use of low temperature chilled water throughout the building, a reduction in the piping and air distribution can be achieved. Other significant advantages of using low temperature chilled water are a reduction in mechanical room space, ceiling space, and electrical installation. Factors that must be considered when designing a low temperature air distribution system when using ice thermal storage systems are reported by (Fields, 1991).

The overall energy consumption of ice storage can increase, due to the production of low temperature chilled water during the night time. During the night time, the chiller must cool a water-glycol solution to a temperature of between -9.0° C to -3.0° C rather than produce a water temperature of 5.0° C to 7.0° C as for conventional systems. This has the effect of reducing the nominal chiller capacity by approximately 30% to 40%. However, compressor efficiency will vary only slightly because lower night time dry bulb temperatures in the case of air cooled chillers, result in cooler condenser temperatures and this will help the chillers to operate more efficiently.

When using ice storage systems, the size of the chilled water pumps is larger than in conventional systems because of the extra head loss resulting from the pressure drop in the ice storage tanks. However, this problem can be overcome by properly designing the storage system with a higher temperature differential. The sizes of the pumps and pipes can be significantly reduced by designing the system with the reduced flow rates that result from using a larger temperature differential in the water loop. Use of a larger temperature range, for example 10.0° C instead of the more traditional 5.5° C temperature range, results in a reduction in the water flow rate and therefore reductions in the pumps and pipe sizes as well as their costs.

There are several types of ice thermal storage systems, and these are briefly discussed below:

3.3.1.1 Ice harvesters

An ice harvester system, as shown in Figure 3.1, consists of an open insulated storage tank. During charging time, ice is built on the vertical plate surfaces of the evaporator, which are positioned above the storage tank. Water is run by a circulating pump at a temperature of 0.0° C and a flow rate of 0.144 $cm^3s^{-1}W_t^{-1}$ to 0.215 $cm^3s^{-1}W_t^{-1}$ cooling on the outer surface of the evaporator, which is fed internally with liquid refrigerant. The thickness of the ice build-up on the evaporator plates ranges from 8 mm to 10 mm depending on the length of the freezing cycle. The ice is harvested by breaking up the supply of liquid refrigerant by feeding hot gas to the evaporator. This raises the temperature of the outer surface to about 5° C, causing the ice in contact with the plates to melt and fall into the storage tank: the build up of ice is stopped by a photo-electric switch when the storage tank is fully charged. During the discharging time, the chilled water from the load is circulated through the storage ice tank further reducing the chilled water temperature to cope with the load.

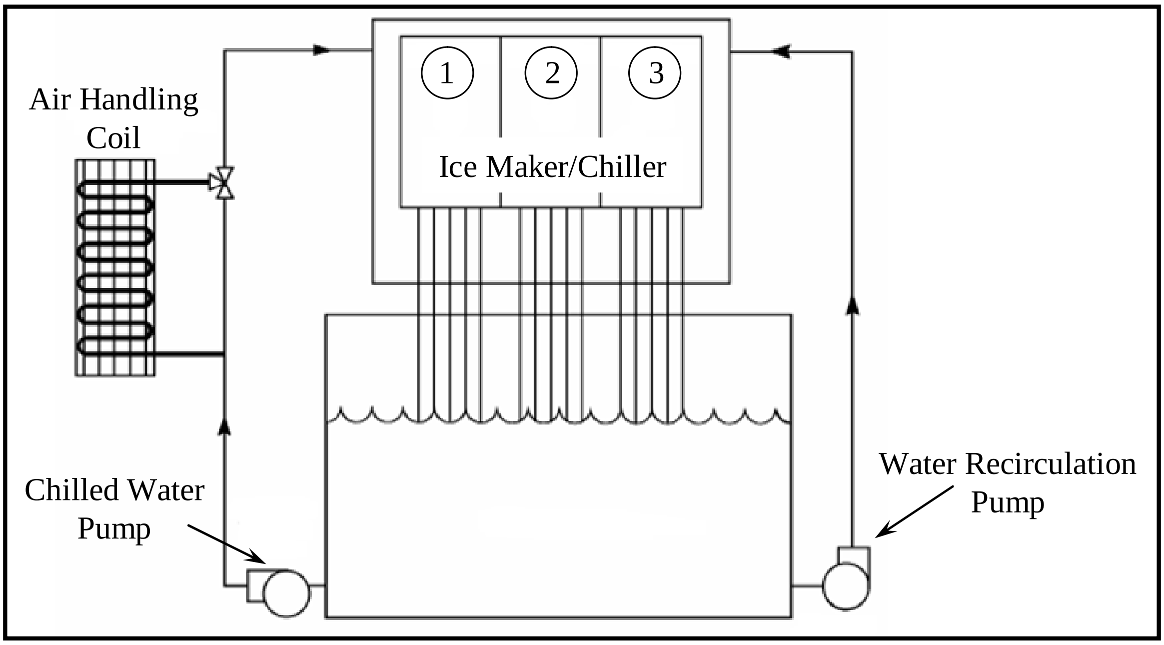

Figure 3.1 Schematic diagram of a typical ice harvesting ice storage system.

3.3.1.2 Ice slurry

In ice slurry storage systems, see Figure 3.2, ice particles are generated by passing a weak glycol/water solution approximately ranged between 2% to 10% glycol by mass through tubing that is surrounded by an evaporating refrigerant contained within a shell and tube heat exchanger. As the glycol/water solution is cooled by the evaporating refrigerant, a suspension of ice crystals is formed. The production of small ice particles within a solution of glycol and water can then be pumped, or can be dropped directly into the storage tank depending on the system configuration.

The ice store discharges by circulating the cool solution from the tank either directly through the AHUs or through an intermediate heat exchanger that isolates the AHUs system from the ice slurry system.

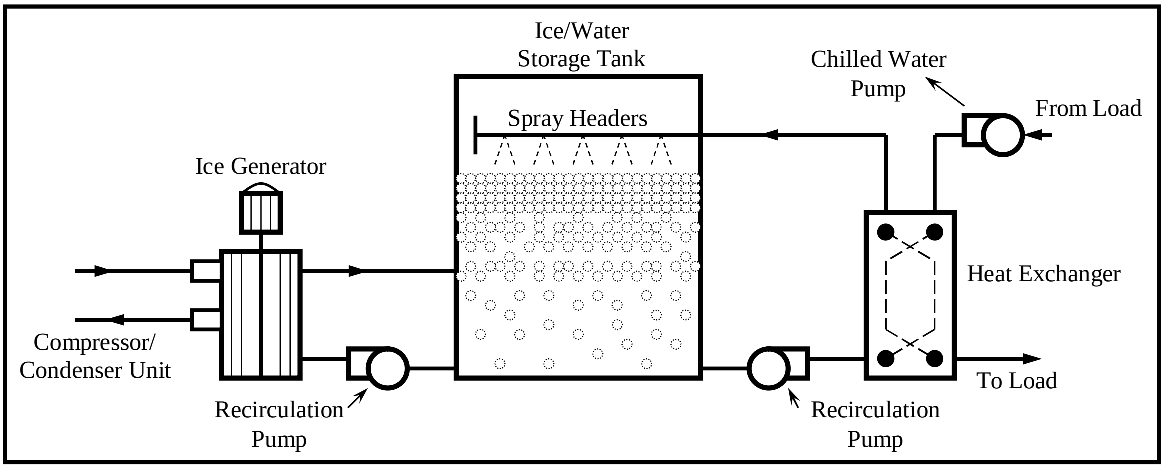

Figure 3.2 Schematic of ice slurry storage.

3.3.1.3 Encapsulated ice

An encapsulated ice storage system consists of spheres or rectangular plastic capsules of water immersed in a secondary coolant such as ethylene glycol in a steel or concrete tank. In the United States, rectangular containers of approximately 0.017 m^3 and 0.0042 m^3 size, and dimpled spheres of 100 mm diameter capsules are available (see Figure 3.3). In Europe, spheres of 95 mm and 75 mm diameters are also used (Dorgan, 1994). The capsules are made of high-density polyethylene and are designed to withstand the pressure due to the expansion during freezing.

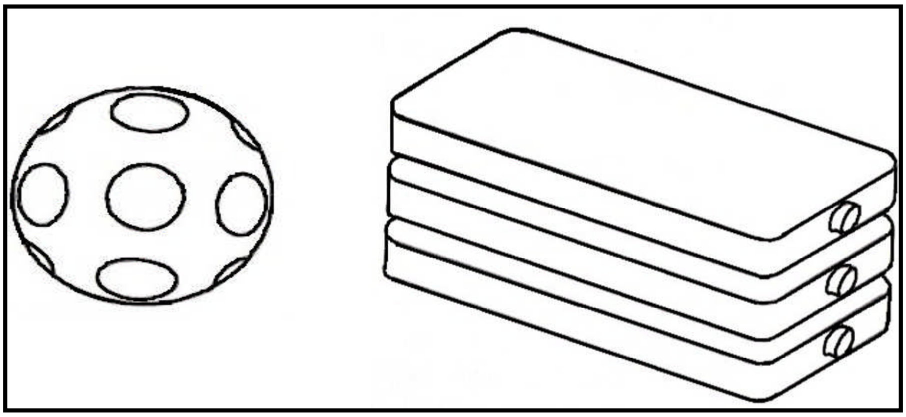

Figure 3.3 Sample of encapsulated ice container.

To charge up the storage tank, a low temperature glycol solution (i.e. from -6.0 to -3.0° C) circulates through the tank causing the water in the capsules to freeze by giving up its latent heat. To discharge the storage, the warm glycol solution returns from the load to the tank and ice in the capsules melts. The charging and discharging of encapsulated ice storage is shown in Figure 3.4.

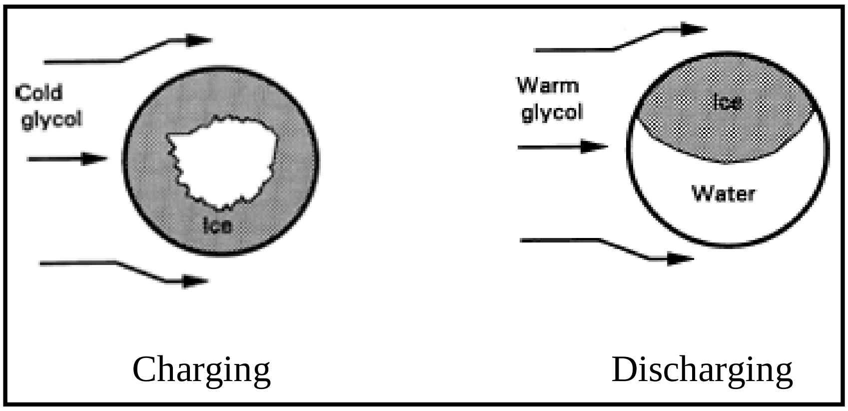

Figure 3.4 Charging and discharging of encapsulated ice storage.

3.3.1.4 External melt-ice-on-coil storage system

The external melt ice-on-coil system is sometimes referred to as an ice builder because in this storage system the ice is formed during the charging time on the outer surface of the heat exchanger coils submerged in an insulated open tank of water as shown in Figure 3.5. The coil is made of steel tubing or schedule 40 welded pipe.

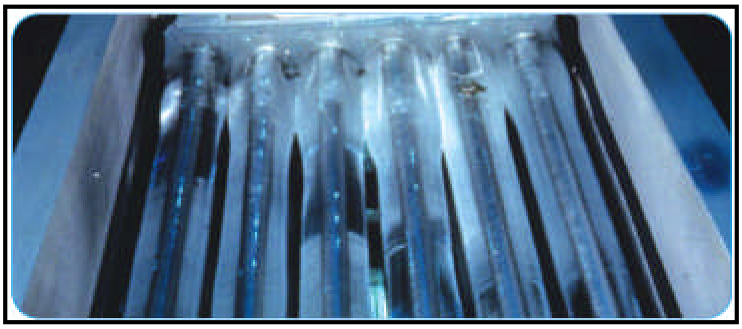

Figure 3.5 Ice formations on the outer surfaces of coils.

During storage charging, a liquid refrigerant or a glycol solution circulates inside the heat exchanger coils, causing ice to form on the outside surface of the coils with a thickness of 40mm to 65mm depending on the application. For applications where higher charging temperatures (i.e. -7.0° C to -3.0° C) and greater efficiency are desired, a thinner layer of ice is formed, and for applications where lower charging temperatures (i.e. -12.0 to -9.0° C) and less efficiency are desired, a thicker layer of ice is formed. During storage discharging, the return water from the load circulates through the ice tank and is cooled by direct contact with the ice. Figure 3.6 shows the charging and discharging processes of external melt ice on coil.

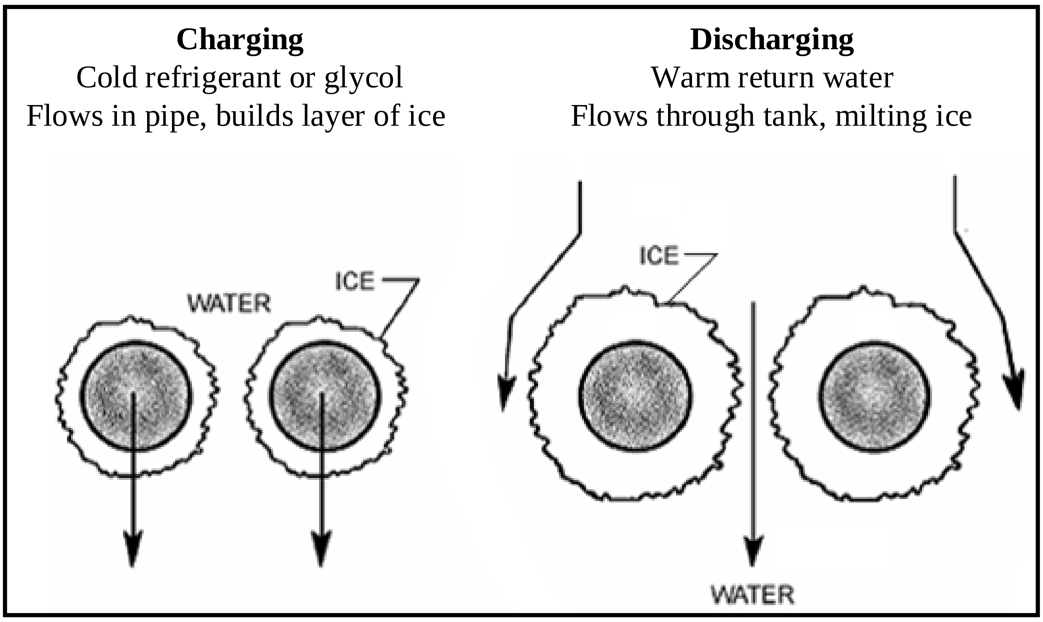

Figure 3.6 Charge and discharge of external melt ice storage.

3.3.1.5 <u>Internal melt –ice-on-coil storage</u>

In the internal melt ice-on-coil storage systems, the heat transfer fluid such as a glycol solution circulates through winding coils submerged in tanks filled with water. During charging, the ice forms on the outer surface of the coil when the glycol solution with temperature of -6.0 to -3.0° C flows through the coils inside the tank. During discharging, the warm glycol solution flows through the coil, melting the ice from the inside out and reducing the temperature of the solution for use in meeting the load. The charging and discharging processes of the internal melt ice on coil are shown in Figure 3.7.

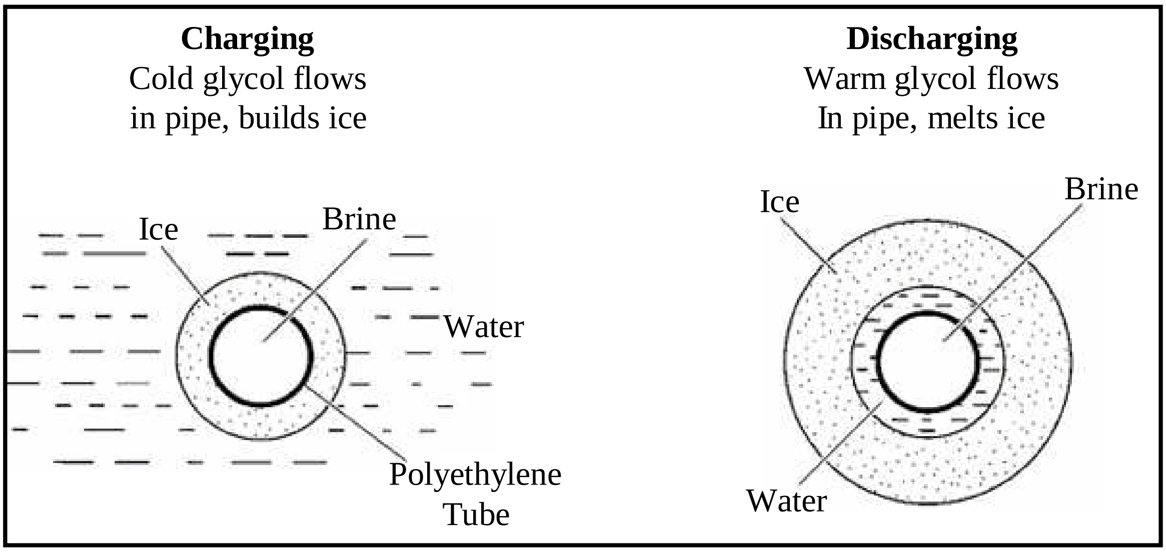

Figure 3.7 Charging and discharging of internal melt ice storage.

3.3.2 Chilled water storage system

Chilled water storage systems are unlike ice storage systems in that they rely completely on the sensible heat capacity of the water and the temperature difference between supplies and return water going to and from the cooling load. As a result, the storage volume required is greater than for any of the ice or eutectic salts storage systems. In the chilled water systems, large storage tanks are used to store chilled water at a temperature between 4.0° C and 6.7° C. This temperature is compatible with most conventional cooling systems and allows the use of a conventional chiller. The chillers in these systems cool the water during the night time when the demand for electricity is low and store it in the tank for later use in the day time period when the demand for electricity is high.

3.3.3 Eutectic salt phase change materials storage systems

The eutectic salt systems are similar to the encapsulated ice systems, but the plastic capsules in the tank contain a eutectic salt instead of water. One type of stacked eutectic salt container is shown in Figure 3.8. A eutectic salt is a chemical mixture of inorganic salts, water, and nucleating and stabilising agents, which melts and freezes at 8.3° C (Dorgan, 1994).

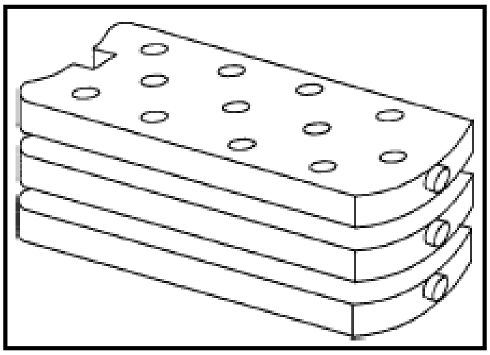

Figure 3.8 Stacked containers of a eutectic salt storage system.

In a eutectic salt system, the storage tank can be charged with a typical conventional chilled water temperature of 4.0° C to 6.0° C, and a standard chiller can be used without the efficiency penalty of a lower evaporator temperature. The stored chilled water can be discharged with typical discharge temperatures of 9.0° C to 10.0° C, which are relatively high for most AC systems and warmer than is normally supplied to the cooling load. This system is generally applicable where humidity control is not a concern.

Further descriptions, control, interface with the building systems, operation, efficiency, and maintenance of the different cool thermal storage systems given above are discussed in (EPRI, 1987; Dorgan, 1994; ASHRAE, 1999b; Seaman, 2000).

3.4 Operation strategies of cool thermal storage systems

Several operation strategies are available for charging and discharging storage to meet cooling load during peak time. The operating strategies of cool thermal storage systems are classified as either full storage or partial storage, referring to the amount of cooling load transferred or shifted from the night to the day period. Partial storage is further classified into a storage priority and a chiller priority.

3.4.1 *Full storage*

Full storage systems are designed to utilise the stored cooling to shift all building cooling loads from the on-peak period to the off-peak period of the design day. In this operation strategy system, the chiller runs at its full capacity during the night period when the building load is small. In the night period, the chiller charges the storage and meets the building cooling loads simultaneously. Since full storage systems meet all the building cooling loads during the day time, this will result in larger and therefore more expensive chiller and storage units compared to partial storage systems.

Full storage systems are likely to be attractive under the following conditions (Dincer, 2002a):

a. Spikes in the peak load curve are of short duration.

b. Time of use energy rates based on short-duration peak periods.

c. There are short overlaps between peak loads and peak energy periods.

d. High peak demand charges apply.

3.4.2 *Partial storage*

In a partial storage system, the chiller operates to meet part of the cooling load and the rest is met by the storage tank during the day time. Usually in this system design, the chiller is sized at a capacity smaller than the design load. Partial storage systems can be further classified based on the selected operation strategies, load levelling, or demand limiting operations.

In a load levelling system, the chiller operates at full capacity for 24 hour of the design day. When the building cooling load is less than the chiller capacity, the excess cooling is stored in the storage tank until the tank is full. When the load exceeds the chiller capacity, the additional cooling is supplied from the storage tank.

The load levelling operation is suitable for applications where the peak cooling load is much higher than the average load (i.e. the ratio of peak to average load is high) and the load is high for a long period. It can be designed to minimise the size, and therefore the cost of both the chiller and storage tank, but it reduces the electricity demand during the day time period less than the full storage system does.

In a demand limiting system, the chiller operates at reduced capacity during the day time. This system design represents the middle ground between full and load levelling partial storage where the chiller operation is reduced. The chiller in this system may be controlled to keep the billing meter at the facility below a given level. The system size, cost, and saving in energy demand fall between that for full storage and that for load levelling partial storage systems. Figure 3.9 illustrates the basic operation strategies of load levelling and demand limiting partial storage, and full storage systems.

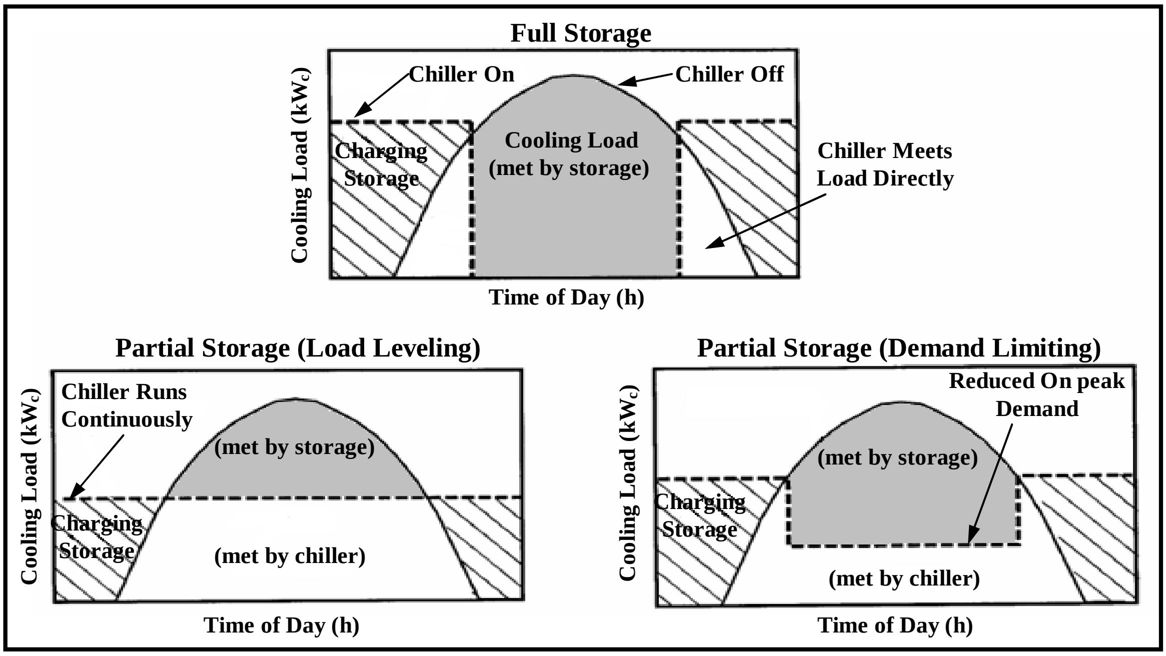

Figure 3.9 Operation strategies of a basic thermal storage system.

Storage priority and chiller priority are two basic operating strategies that can be applied when the system is designed for partial storage. These operating strategies divide the load between the chiller and the storage. The storage priority strategy meets as much of the load as possible from the storage tank, using the chiller only when the load exceeds the total stored cooling capacity of the tank. However, the chiller priority strategy uses the chiller to meet as much of the load as possible. Cooling is supplied from the storage tank only when the load exceeds chiller capacity. Storage priority generally has a more complex control scheme and larger storage and chiller sizes than the chiller priority strategy. However, it consumes less energy (Simmonds, 1994).

3.5 Selection of appropriate cool thermal storage types for Kuwait

In the market, there are many ice cool thermal storage technologies available; each storage technology has its own advantages and disadvantages, and examining in depth the suitability of the individual system for Kuwaiti conditions is a very lengthy and difficult process because their performances, controls, and piping configurations are different. However, particularly for ice thermal storage systems, there is some technical information that is common to Kuwait and other countries such as the United Kingdom and the United States that can be considered in order to make a suitable selection for Kuwaiti conditions. A literature survey was conducted into the costs, operations, and applications and customer satisfactions of incorporating different ice storage systems in the United States (Dorgan, 1994; Potter, 1995; Shan, 2001). It was found that ice harvester and ice slurry are the most expensive systems compared to other ice storage technologies. According to (Dorgan, 1994), the cost of the ice harvester chiller system ranges from 313 US$ per kW_t to 427 US$ per kW_t compared to the range of 57 US$ per kW_t to 142 US$ per kW_t for other ice storage technologies. An ice slurry system is similar to an ice harvester system and both have limited commercial application because of their higher cost (Shan, 2001); therefore, both systems may not be fit for buildings in Kuwait because these systems greatly raise the capital cost of the AC.

Storage interface with the building is another factor must be considered in selecting ice storage technology. All types of ice thermal storage systems except internal melt ice-on-coil and possibly encapsulated ice systems use open non-pressurised tanks. Open non-pressurised storage tanks are highly aerated to the external environment so water treatment, make up water, and corrosion protection may be of increased concern especially in very hot countries such as Kuwait and where the sand storm in the summer period may last for about two months, as well as the cost of make up water being very high because of the need to rely on desalination for water production. Moreover, open tanks are always associated with extra valves, transfer pumps, and heat exchangers depending on the system design. Pressure sustaining valves and automatic positive shut off valves are required to control the flow rate in and out of the open storage tanks, transfer pumps are required to circulate chilled water from the open storage tanks to the closed and pressurised chilled water system in the building, and in some designs, a heat exchanger is added to isolate the non-pressurised open circuit from the pressurised building circuit. Adding pumps, values, and a heat exchanger will result in an increase of initial and maintenance costs of the cooling system particularly for small buildings such as this clinic building.

Internal melt ice-on-coil systems and encapsulated ice systems have the same features, chiller and storage costs, charging and discharging temperatures, chiller charging efficiency and so on. So one of two types of ice thermal storage technology was selected and examined for the clinic building. Selecting an internal melt ice-on-coil storage system was appropriate because its performance data, piping, control, and specifications were available in the market of Kuwait and could be obtained particularly from Awali General Trading and Contracting Company, the representative of CALMAC ice banks in the U.S and Fawaz Refrigeration and Air Conditioning Company, the representative of Dunham-Bush Ice-Cell in the US.

Furthermore, according to Potter (1995), a survey of 196 thermal storage systems was conducted. The users of the thermal storage systems were asked to assess the ice storage systems according to an index of 1 to 10. It has been found the internal melt ice-on-coil system had the highest rating of 7.82 compared to other ice storage systems.

Although chilled water storage system technology is suitable for buildings that have large cooling loads, it may be possible to study its application for small buildings such as the clinic. Using a chilled water storage system requires much smaller chillers compared to other storage technologies, and this may offset the extra cost associated with the storage tank. Most large chilled water storages use open systems; however, since the clinic building is small, a smaller storage pressurised tank may be used to eliminate the need for extra transfer pumps and pressure sustaining values. According to Dorgan (1994), cylindrical pressurised steel tanks in sizes of 11 m^3 to 210 m^3 can be used for chilled water storage.

The eutectic salt storage system may not be suitable for the clinic because of the higher typical discharge temperature, which ranges from 9.0° C to 10.0° C and because the cost of the storage tank is higher than in other storage system technologies. Furthermore, it uses open non-pressurised tanks, which further increases the capital cost of the system. Since an internal melt ice-on-coil and chilled water storage system was selected to be examined for the clinic building, further discussion will be given of their operation, control, and design in the following subsections.

3.6 Internal melt ice-on-coil storage system

3.6.1 Configuration and control strategies

In an internal melt ice-on-coil storage system, a chiller and ice storage tank can be arranged in series, and the chiller can be configured either upstream or downstream of the storage tank as shown in Figure 3.10 (a) and Figure 3.10 (b) respectively.

In the chiller upstream configuration, the return chilled water from the AHUs is first precooled by the chiller before entering the storage tanks as shown in Figure 3.10 (a). This arrangement offers the advantage of running the chiller with the highest possible operation temperatures and therefore, at its best coefficient of performance. Furthermore, in this arrangement, the chiller has a greater capacity at higher temperatures.

Alternatively, in a series arrangement, the chiller can be arranged downstream of the ice tank as shown in Figure 3.10 (b). The return water from the AHUs first flows through the ice tanks and cools before entering the chiller. This has the advantage of a high usable storage capacity, as well as an assured constant discharge temperature. However, the chiller operates at the low temperature condition; therefore, the coefficient of performance of the chiller in this configuration is low.

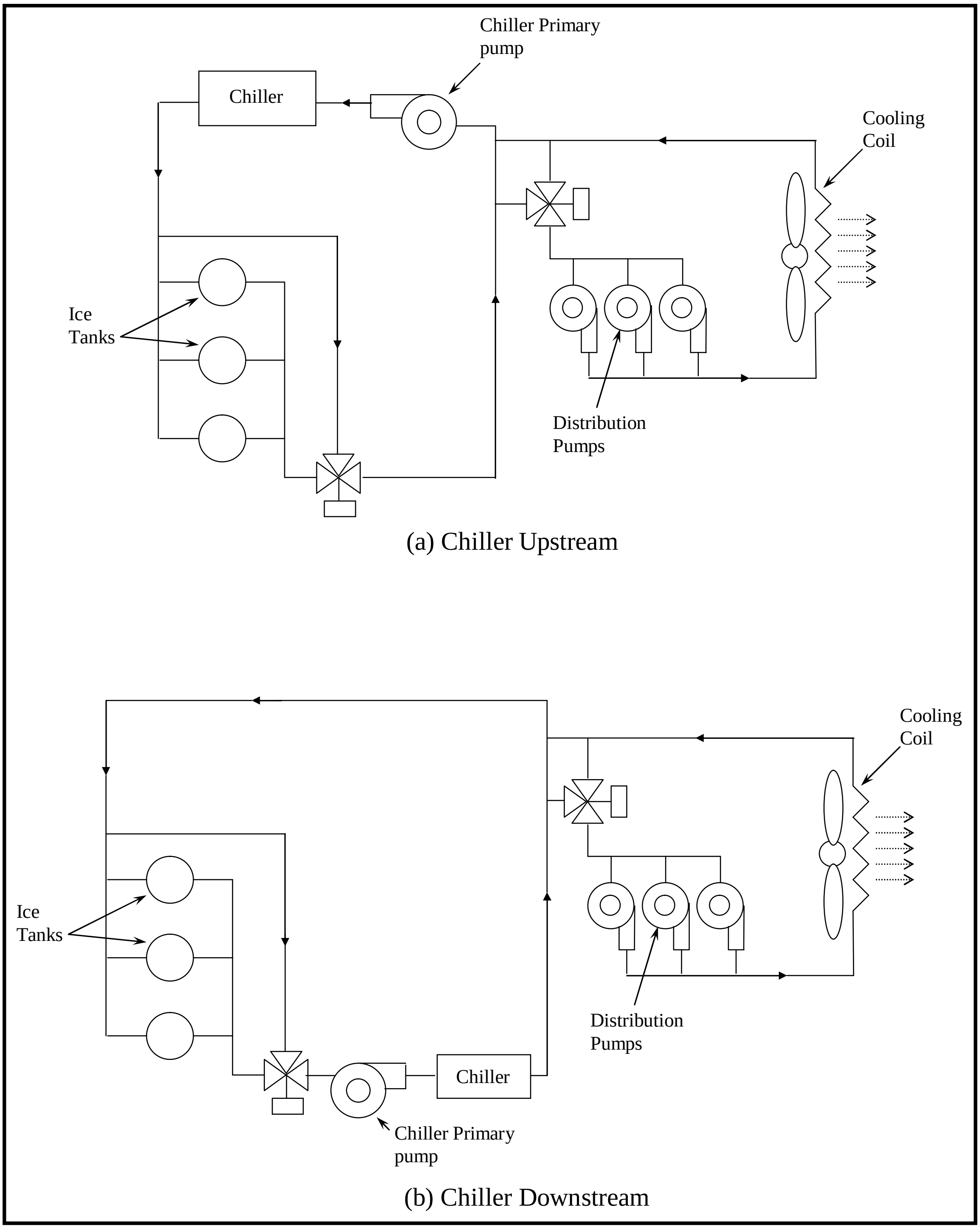

Figure 3.10 Series flow with primary and secondary pumping circuits.

A Parallel configuration of ice tanks and chiller as shown in Figure 3.11 is an alternative configuration suitable for lower temperature differentials across the cooling circuit of less than 7.0˚ C (CIBSE, 1994). In this arrangement, both chiller and ice tanks receive the benefit of high return chilled water from the AHUs. As a result, the chiller operates at high capacity as well as having a high coefficient of performance, and the net usable portion of the storage capacity can be maximised. However, the main disadvantage of this arrangement is that the piping configuration is more complex than the series arrangement.

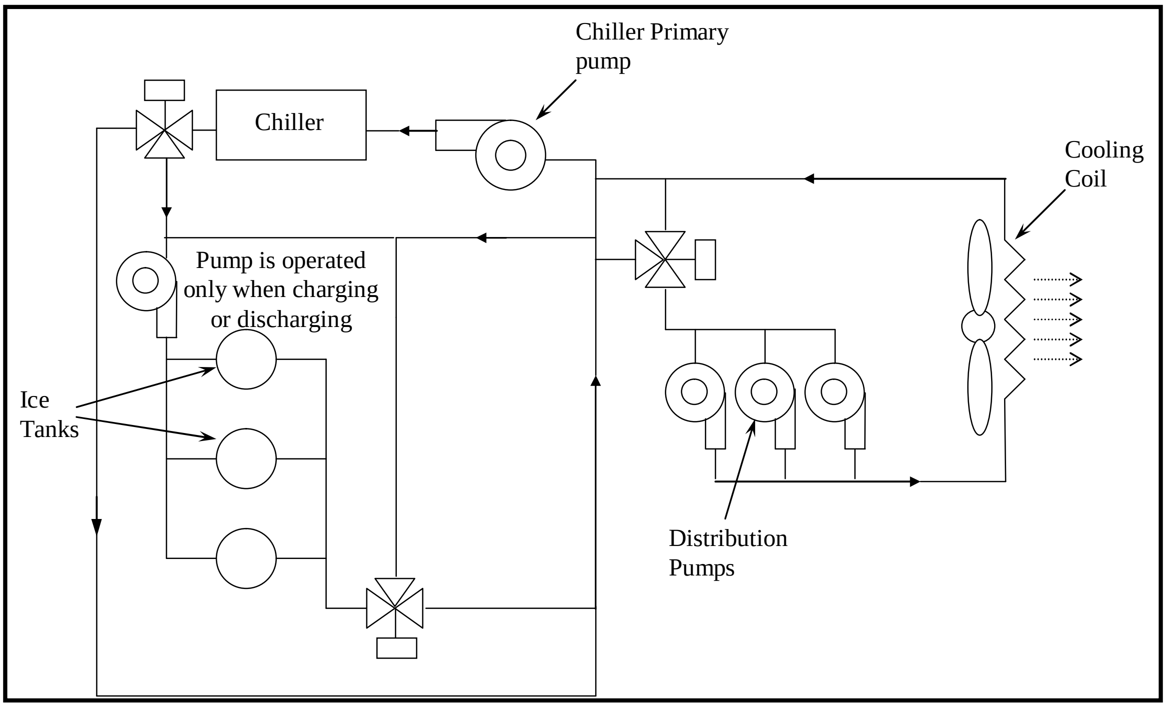

Figure 3.11 Parallel flow with primary and secondary circuits.

3.6.2 *Energy Performance*

3.6.2.1 The energy performance of the chiller

As discussed in Section 3.3, the chiller in an internal melt system must be able to produce a chilled water temperature of -6.0° C to -3.0° C during the charging time, which means the chiller in this system requires an evaporating temperature several degrees below 0.0° C. This compares with the evaporating temperature above 2.0° C for a conventional system. As a result, the coefficient of performance of the chiller in the ice storage system during the charging mode drops. However, this situation should be partially offset since the charging mode always happens during the night time, so the chiller takes advantage of the lower ambient temperatures and operates at a reduced condensing temperature, hence, the higher coefficient of performance.

The coefficient of performance of a chiller is also affected by the piping configuration. For an upstream arrangement chiller, the chiller will have a high coefficient of performance because of the high return chilled water entering the chiller during discharging mode but the low usable percentage of storage capacity. For the downstream chiller arrangement, the usable portion of storage capacity is maximised; however, the coefficient of performance of the chiller is minimised because of the low operating temperature.

3.6.2.2 The energy performance of the ice storage tank

The energy performance of an ice storage tank depends on the thermal losses from the storage container. The heat transfer to an ice storage tank is a function of the total surface area of the tank, the heat transfer coefficient of the tank walls and the temperature difference between ice in the storage tank and the surrounding.

The internal melt ice storage tanks are typically delivered as prefabricated modular tanks (Dunham-Bush, 1995; CALMAC, 2001b) as shown in Figure 3.12, and may be constructed of fibreglass or high density polyethylene. Furthermore, these tanks can be partially or fully buried, and can be stacked indoors or outdoors. For fully or partially buried tanks, the heat transfer properties of the surrounding soil must be estimated. For tanks located outdoors, the tanks are commonly exposed to sunlight and to radiation heat transfer, and this may increase the heat transfer to the tank and raise the water temperature. Therefore, the tanks are well insulated and covered with a vapour barrier, and are weatherproofed to protect the external insulation. To reduce the heat transfer to the ice from the surroundings, the ice tanks are typically insulated with urethane foam insulation (Dunham-Bush, 1995).

Figure 3.12 Prefabricated modular internal melt ice on coil tanks.

According to Dorgan (1994), the thermal conduction losses from storage tanks are typically in the range of 1% to 5% of storage capacity per day. In the case of the ice storage tanks, the standby losses must not exceed 1% of the total storage capacity per day under a constant ambient air temperature of 29.4° C (CALMAC, 2001b). Dunham-Bush provides ice tanks with 0.8% loss of total storage capacity per day under 32.0° C ambient air.

3.6.2.3 Energy performance of the pumps

Another factor that may affect the overall energy performance of the internal melt ice-on-coil is the extra pressure drop that is associated with the presence of the storage tank. This extra drop in pressure increases the power consumption of the chilled water pumps in the system. Based on performance data (CALMAC, 2001a), an ice storage tank could have a pressure drop between 9.8 kPa m to 172.5 kPa depending on its size and design flow rate. The overall extra pumping power resulting from the pressure drop in the tanks may not be so significant because ice storage systems are usually designed based on high temperature differential results in a low flow rate and hence, less pumping power.

3.7 Chilled water storage system

In a chilled water storage system, the amount of cooling energy that can be stored in a tank depends on the temperature difference between the return warm water from the AHUs, the temperature at which the water is stored, and the amount of chilled water stored in the tank. The cooling capacity of the storage tank can be maximised by maintaining a high chilled water temperature differential. In order to maximise this temperature differential, the water temperature exiting the AHUs must be maximised, the stored water temperature in the tank must be minimised, and mixing of the warm return water and stored chilled water in the tank must be prevented as much as possible.

The cooling capacity of the storage tank can be reduced during the storage process of chilled water by (Shan, 2001)

a. Increasing the temperature of the stored chilled water by direct mixing of warmer return and stored chilled water.

b. Increasing the temperature of the stored chilled water by heat transfer from the tank wall from the warmer ambient air.

Thermal separation between warm return water from the AHU and cold charged water can be achieved by stratification, multiple tanks, membranes, diaphragms, and baffles (Mackie, 1988; Dorgan, 1994). Dorgan (1994) argued that the stratified tank is the simplest, most efficient, and most cost effective of chilled water storage systems and is very attractive for large buildings and district cooling applications where the storage capacity is greater than 7000 kW_th or about 760 m^3. The larger the tank, the lower the surface to volume ratio, and the lower the cost per kW_th of stored cooling energy.

The chilled water storage tanks are usually flat bottomed rectangular container or vertical cylinders. Cylindrical tanks have a lower surface to volume ratio than have rectangular tanks. Steel is the commonly used material for above ground tanks, and concrete is widely used for underground tanks.

3.7.1 *Configuration and control strategies of the chilled water storage system*

The chilled water storage system can be well designed using the basic operation strategies discussed in Section 3.4, full storage, partial storage load levelling, and partial storage demand limiting. Chiller and storage priority controls can be implemented with the chilled water storage system. Storage priority can be used to optimise a part load operation because the cost of the direct cooling is often higher than the cost of the stored cooling. However, storage priority requires some type of load prediction for the following day. Furthermore, chiller priority control can be also implemented.

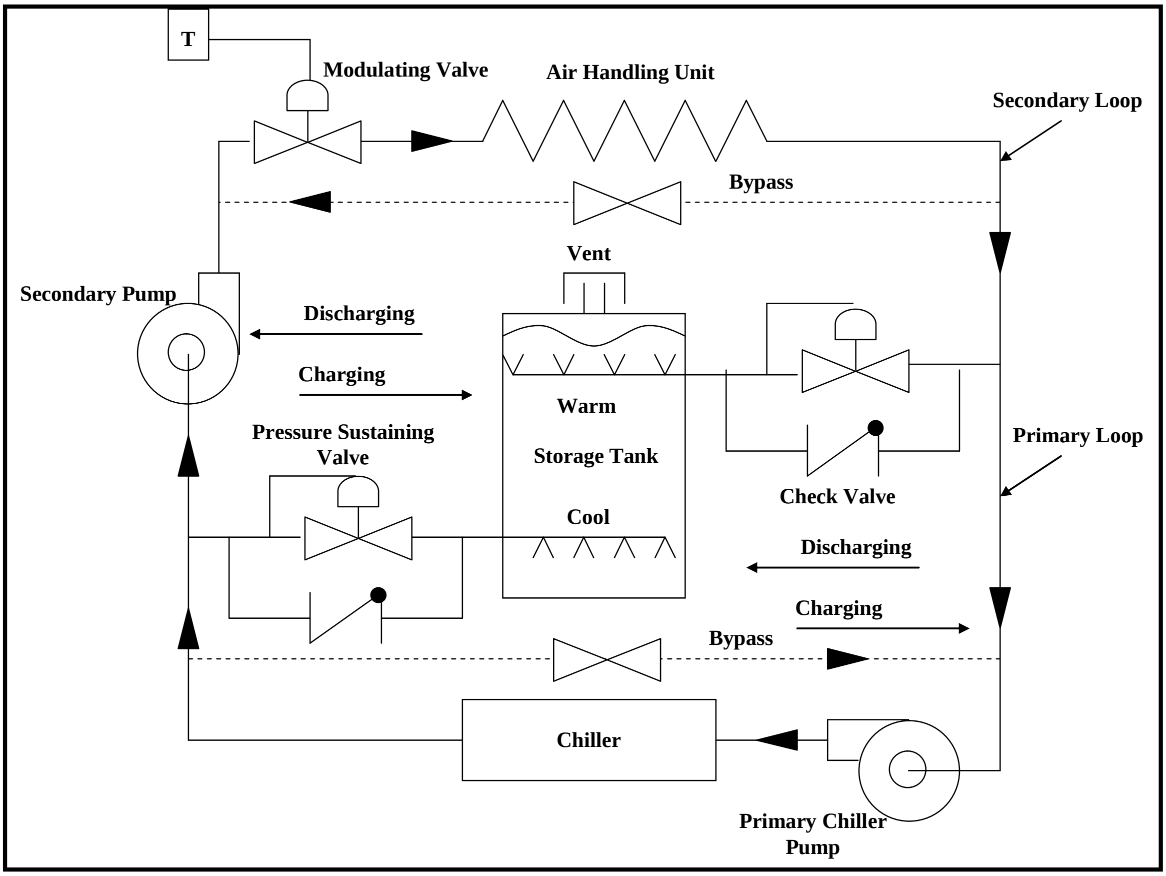

Figure 3.13 Stratified chilled water storage system configuration.

A schematic diagram of a basic chilled water system that can be incorporated into an AC system is shown in Figure 3.13 (Mackie, 1988). The system supplies cooling to the AHU and the secondary distribution pumps supply a variable flow of chilled water to the AHU. When the load is smaller than the chiller capacity, the supplied chilled water comes only from the primary chiller loop.

When the load exceeds the chiller capacity, the chilled water comes from both the chiller loop and storage tank (partial storage demand limiting and load levelling operation strategies), or from only the storage tank (full storage operation strategy). In the charging mode, the primary chilled water pumps supply chilled water to the bottom of the tank and to the secondary distribution circuit, and the return water to the chiller comes from either the top of the storage tank, or from the mixture of the return water from the secondary loop and storage tank. The function of the bypass valve on the chiller in the primary loop is to maintain a constant flow rate of chilled water to the chiller, and the bypass valve located in the secondary loop is used to maintain a constant supply temperature (i.e. at 7.2° C) to the AHU.

3.7.2 Energy performance of the chilled water storage system

The energy performance of a chilled water system depends on storage losses, interface pumping energy, and chiller performance during the charging and discharging operation. AC incorporating chilled water storage may have higher energy consumption than a conventional system due to the internal and external storage heat transfer, the presence of transfer pumps, and a higher chiller energy penalty because of the lower temperature of the chilled water exiting the chiller.

Storage losses cannot be eliminated from the system. However, they can be minimised by proper diffuser design and good insulation of the storage tank. Pumping energy can either increase or decrease depending on the design of the piping system and system temperature differential. Furthermore, chiller efficiency penalty can either increase or decrease depending on the selected exit temperature of the chilled water and the dry bulb temperature for an air cooled chiller or the wet bulb temperature for a water cooled chiller at night time, when the chiller is charging the storage tank.

3.7.2.1 The energy performance of the storage tank

The measurement of the thermal losses in the cooling capacity of the stored chilled water during the charging and discharging processes in a complete storage cycle is indicated by an enthalpy-based figure of merit. 'Figure of merit' is defined as the ratio of cooling capacity available during the discharge process to the theoretical cooling capacity available during the charging process: the smaller the losses of cooling capacity during the discharging cycle, the greater the figure of merit.

Storage efficiency is another way of measuring the losses in the cooling capacity of the stored chilled water, and it is defined as the ratio of the amount of energy removed from storage to the amount of energy put into storage during a complete charge and discharge cycle (Dorgan, 1994). The advantage of using a figure of merit rather than storage efficiency in measuring the performance of a storage tank is that the figure of merit takes into account the actual losses in the cooling capacity due to thermal conduction across the boundaries and the mixing of warm and chilled water in the tank. However, storage efficiency accounts only for the losses in cooling energy from the surroundings and it does not account for the thermal losses by the mixing of the chilled water within the storage tank.

There are two main heat transfer mechanisms that can increase the thermal losses of the storage tank: heat transfer from the surroundings to the stored chilled water in the tank across the tank boundaries, and heat transfer from warm water to the stored chilled water within the storage during the charging and discharging modes. The two basic mechanisms are further discussed in the following subsections.

i. Heat transfer within the chilled water storage tank

Within the storage tank, heat can be transferred to the stored chilled water by the interactions between the stored chilled water and the inner surfaces of the tank, and by mixing and conduction at the interface of warm and stored chilled water. These two basic heat transfer mechanisms can reduce the cooling capacity of the storage tank since they increase the stored chilled water temperature that is not usable for cooling the building.

The interactions between the stored water and the inner surfaces of the tank occur when the interior surfaces of the tank are alternately warmed and cooled by direct contact with the warm and cooled water during the charging and discharging times. For example, at the beginning of the charging process, the temperature of the incoming water to the bottom of the tank is lower than the temperature of the water stored in the tank. Since the temperature of the internal surfaces of the tank is approximately equal to the temperature of the initially stored water mass with which it is in contact, the heat transfers from the internal tank surfaces to the incoming chilled water. In order to reduce this effect, the surface volume ratio of the storage tank must be minimised.

The mixing and conduction at the interface of warm water and stored chilled water is the most significant heat transfer type that occurs in a storage tank. Two important mixing mechanisms occur in a stratified chilled water tank (Wildin, 1990). The first mixing occurs at the start of the charging and discharging cycles during the formation of the thermocline between the warmer water and the cooler water. The initial thickness of the thermocline is largely determined by this mixing process. The thickness of the thermocline increases by heat conduction through the thermocline during the charging and discharging cycles and this disturbs the initial thickness produced by the mixing, and increases heat transfer during thermocline formation.

The second mixing occurs on the inlet side of the thermocline after the thermocline has formed, and is associated with vortices. This type of mixing happens due to the inertia of the incoming fluid, which produces vortices in the chilled water in the tank between the inlet diffuser and the thermocline. This mixing process significantly affects the performance of the tank when it occurs during the charging cycle. In order to reduce the mixing of warm and cold water in a stratified chilled water tank during the charging and discharging cycles the diffusers must be properly designed.

ii. Heat transfer across storage tank boundaries

There are many factors that increase the heat transfer to the stored chilled water in a tank from the surroundings such as solar heat gain through the storage tank, heat gain from the warmer surrounding air, the condensation of water vapour in the air surrounding the tank, and in the case of underground tanks, transfer of heat to the stored chilled water from the temperature of the soil surrounding the tank.

Storage tanks are commonly constructed from welded steel, precast prestressed concrete, and cast-in-place concrete. Steel tanks have higher thermal conductivity than have concrete tanks, therefore they conduct more heat to the stored chilled water especially for tank volumes less than 190 m^3 (Dorgan, 1994). A common problem associated with cast-in-place concrete storage tanks is water leakage from the tank (William, 2003).

For buried tanks, soil thermal conductivity surrounding the tank and the depth at which the tank is buried are two important factors that affect the heat transfer to the stored water. Thermal conductivity of the soil increases with the moisture content. For example, sand thermal conductivity with less than 4% moisture content by mass is 0.29 $Wm^{-1}K^{-1}$, and when moisture content ranges from 4% to 20%, the thermal conductivity is 1.87 $Wm^{-1}K^{-1}$. With moisture greater than 20% by mass the thermal conductivity increases to 2.16 $Wm^{-1}K^{-1}$ (ASHRAE, 2000c).

In some countries such as Australia (William, 2003), it is not common practice to insulate a buried concrete tank. However, the conditions in Kuwait are different; buried steel and concrete tanks must be well insulated because of the high soil temperature. Based on average values obtained from the annual climatological reports of 1970, 1972, 1985, and 1986, the average soil temperature in the peak summer months of June, July, August, and September at a depth of 1.5 m is 34.3° C and at a depth of 3 m is 30.4° C. Moreover, the outside surrounding dry bulb temperature for most of the time in the peak summer period is above 40° C and sometimes reaches 52° C. Therefore, the heat gain to the stored chilled water should be minimised by proper insulation of the storage tanks, using a low ratio of tank surface to water volume and reducing the residence time of the stored chilled water in the tank.

Details of the construction of a 10,161 m^3 stratified chilled water tank to reduce the heat gain from the surrounding boundary are discussed by (Fiorino, 1994). In this construction, the solar heat gain is minimised by a white reflective urethane coating, and heat conduction from the surroundings is reduced using polyurethane insulation. The condensation of water vapour is eliminated by applying a butyl rubber vapour barrier between the polyurethane foam insulation and the white urethane outer coating.

3.7.2.2 Transfer pump energy use

Some chilled water storage systems consume more pumping energy than do conventional AC systems due to the presence of transfer pumps. The purpose of the transfer pumps is to circulate chilled water from the storage tank (open system) to the closed and pressurised chilled water system in the building. The transfer pumps are sized to take into account the extra pressure drop resulting from additional strainers, valves, and diffusers associated with the storage tank, and more importantly the hydrostatic head difference between the highest point in the building system and the water level of the stored chilled water in the tank. In conventional AC systems, the chilled water is transferred directly from the chiller to the building; hence, the pumps in conventional systems consume less energy.

Sometimes the additional energy required for the transfer pumps could be small because in chilled water storage systems, the systems are usually designed with a higher temperature differential than conventional systems are; this means the systems have lower flow rates and hence lower pumping power requirements.

Transfer pump interfaces are divided into two classes: indirect and direct. An indirect pumping interface physically isolates the chilled water stored in the tank from the rest of the chilled water system by using a heat exchanger. Direct pumping interfaces allow water to flow between the open storage tanks and pressurised chilled water system. Direct interfaces are further classified based on whether they incorporate hydraulic energy recovery and whether they are unidirectional or reversible. These types of interfaces are ably described by William (1999).

Transfer pumping energy is minimised in chilled water storage systems by using variable speed pumps or insulating the open tank from the closed chilled water system by a heat exchanger, or by using a pressure recovery pump, which can be derived from a similar pump equipped with a turbine wheel (James, 1996; William, 1999).

Placing heat exchangers between the storage tank and the chilled water system circuit creates independent hydraulic circuits, each with its own pumps, piping, and control. This arrangement reduces pressure drop, simplifies system design, reduces pump motor size, and eliminates the hydrostatic pressure head that exists between the storage and building system, thus minimising the energy consumption of the pumps. This technique can be attractive for a high hydrostatic head difference. However, the disadvantage of using heat exchangers with a storage tank is that they reduce the temperature differential available for storage, which means increasing the size and cost and reducing the cooling output of the storage tank. The design temperature differential between the cool inlet and warm outlet temperatures is typically no less than 1.1° C. When a heat exchanger is used, the temperature differential may raise the temperature differential to about 2.0° C resulting in a higher temperature loss on both charge and discharge cycles (William, 2003).

Using direct rather than indirect pump interfaces maximises the temperature differential across the storage tank, thus decreasing the tank size and increasing the storage cooling output. Another advantage of using direct pump interfaces is the lower capital and maintenance costs because they require less equipment, fewer controls, and fewer valves. However, in some applications such as district cooling plants or high rise buildings, and when the height of the building is much greater than that of the storage tank, direct pump interfaces may not be attractive, because of the increase in the sizes of pumps and pressure sustaining valves as well as the significant increase in pumping energy.

In high static pressure applications, hydraulic turbine energy recovery can be used for the direct pumping interface. An energy recovery turbine in the return chilled water line to the tank is connected to a transfer pump through a double extended motor shaft. With a pressure sustaining valve, the turbine operates in series to absorb excess system pressure as water flows back to the storage tank, converting the energy associated with the static pressure differential into shaft power that directly reduces pump motor input power. In typical applications, a turbine can recover 35% to 50% of the peak pump motor power (William, 1999).

3.7.2.3 __Chiller energy performance__

The energy consumption of the chillers in a chilled water storage system can be significantly reduced, because the chillers operate with a higher coefficient of performance and at a reduced condenser temperature. The energy consumption of a chiller increases as the chiller load or the inlet condensing water or dry bulb temperature increases. Since storage charging usually takes place at night time, when the wet bulb or dry bulb temperature is low, the water cooled or air cooled chiller can be operated with a lower inlet condenser temperature, which consequently makes the chiller more efficient.

Another factor effecting the energy consumption and the coefficient of performance of a chiller is the selected design temperature of the water exiting chiller. For example, for a chilled water storage system, if the chiller is to supply chilled water at a temperature of 4.0° C to 5.0° C, the energy consumption will be slightly higher and the coefficient of performance lower than for a conventional system where the supply chilled water is in the range of 5.5° C to 7.0° C.

3.7.3 The design of the diffusers for the chilled water tank

Diffusers are the most important element in the successful performance of any stratified chilled water storage system. The diffusers are a specially designed pipe network, and are usually installed at the top and bottom of the tank, with the upper diffuser mounted beneath the surface of the water, and the bottom diffuser fixed above the tank floor. The main purpose of the diffusers in the storage tank is to allow water to enter and leave the tank gently without causing significant mixing between warm water and chilled water during the charging and discharging cycle; they also form and maintain a thermocline and then ensure that the thermocline is not disturbed by any subsequent mixing. A thermocline is defined as a stratified region in which there is a steep temperature gradient; the water in this region often varies from 6.0° C to 16.0° C. In a good chilled water storage design, the thickness of the thermocline is about 0.3 m to 1.0 m (Dorgan, 1994). It is essential in the chilled water storage design to keep the thermocline very thin in order to reduce the mixing loss of the tank.

Diffusers are installed in different shapes based on the shapes of the storage tanks. For example, for cylindrical tanks, octagonal and radial disk diffusers are best suited, and for square and rectangular tanks, the best diffusers are of horizontal slot and H styles (Mackie, 1988; Dorgan, 1994). Figure 3.14 shows two types of diffuser arrangement.

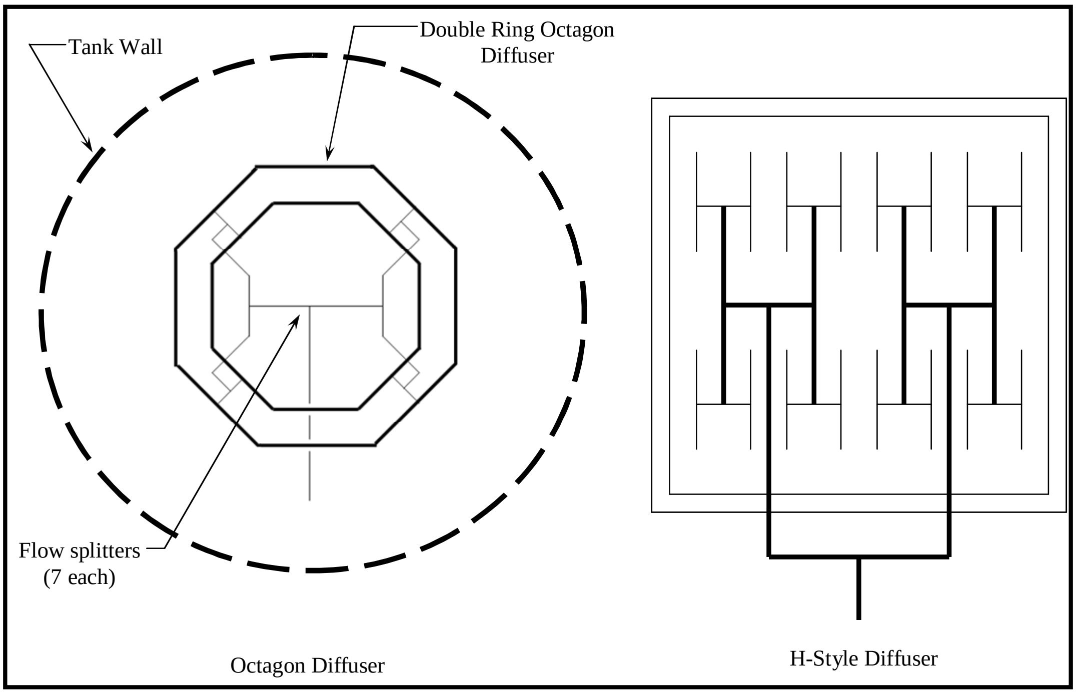

Figure 3.14 Octagon and H Style diffusers arrangement.

The principle and procedures of diffuser design are well discussed by (Wildin, 1990). The design is valid for charging temperatures down to 4.0° C. For lower temperatures, the diffuser design reported by (Wildin, 1990) is not applicable since this study is based on the assumption that the density of the chilled water in the storage tank increases as the temperature decreases.

Further discussion of diffuser designs for single and double octagon diffusers is reported by (Fiorino, 1991). Fiorino (1991) demonstrated that because of the large height to diameter ratio of 0.39, a double octagon diffuser was selected instead of a single octagon diffuser for the design of the storage tank to increase the effective diffuser length and therefore reduce the flow rate per diffuser length.

Four ring diffusers were installed in an 8,517 m^3 chilled water storage system for a central energy plant at an army installation in the US (Sohn, 1999). The diffuser system in the storage tank as reported by (Sohn, 1999) has performed very well. The diffuser's performance was determined from the measurement of the thermocline movement inside the storage tank during the charging process, which ranged from 0.3 m to 0.61 m;

the useful tank volume lost due to the thermocline was only 5%. Further detailed discussion of the diffuser design and construction reported by (Sohn, 1999) is illustrated in the study by (Sohn, 1998).

3.8 Case studies

To further understand the advantages of using cool thermal storage systems to time shift the electrical energy consumption and to reduce the power demand of the AC system, the following case studies for some existing successful implementations are discussed.

3.8.1 Ice thermal storage systems

3.8.1.1 Office building, Dallas, Texas US

An internal ice-on-coil storage system was installed in an office building located outside Dallas, Texas US to shift about 800 kW_e electric demands from the day time to the night time, and save an estimated US$ 55 thousand each year in demand charges (Tackett, 1989). Before the project was implemented, the AC design team developed and studied four alternate different cooling systems schemes: conventional, load levelling partial storage, demand limiting partial storage, and full storage systems. For ice storage system designs, the team modified and optimised construction costs, utility savings, and operating flexibility, and compared equipment sizes, incremental costs, and savings with those of the conventional system.

The team found that the load levelling partial storage system strategy had the smallest chiller and storage sizes, provided the least savings, and had limited flexibility because it could not afford chiller shut down or operate without the ice storage. Furthermore, the full storage achieved maximum savings but had bigger storage and the largest chillers, and could provide the greatest flexibility since it could operate as a full storage, a partial storage, and as a conventional system without the use of the storage tank. In the demand limiting partial storage system, the designers found that the chiller and storage sizes fell somewhere between those of load levelling partial and full storage systems, and the overall costs and saving were optimised.

The above system can be compared to the conventional system. Tackett (1989) reports that the ice thermal storage system has a simple payback period of 0.34 years for load levelling partial storage, 2.35 years for demand limiting partial storage and 1.34 years for full storage systems. Furthermore, load levelling storage has the lowest initial cost; full storage has the highest, and the initial cost of a demand limiting system falls between the two systems. Based on their design and cost analysis, the team selected a demand limiting operation cooling system to cool the building.

The cooling system of the building consists of two glycol three-stage centrifugal water cooled chillers each of 2.0 MW_t and 90 internal melt ice-on-coil tanks connected in parallel as well as a heat recovery glycol chiller, which has a double bundle condenser and operates as a heat recovery chiller in the winter. In peak summer period, the system is operated in a three step sequence to reduce demand charges during the utility's on peak period:

a. At 20:00 hours, both glycol chillers produce chilled water at a temperature of -4.4° C to charge 90 ice tanks, the chilled water circulates in the tubes inside the tanks, and freezes the water surrounding the tubes, and returns at a temperature about -1.1° C.

b. By 6:00 hours or 7:00 hours, the tanks are completely charged. The temperature of chilled water exiting the chiller is reset to 1.1° C, and is then pumped to the AHUs in the building for cooling.

c. At about 12:00 hours to just before 20:00 hours, one of the chillers is switched off. The second chiller is controlled to operate at a reduced demand limit. This instantly lowers the on-peak electrical demand of the chillers and thus reduces the demand charges. Then, after 20:00 hours, the process is repeated.

To improve the efficiency of making low temperature chilled water, the water cooled chillers in the central plant use three three-stage centrifugal compressors. Each chiller provides 2.0 MW_t of cooling at a coefficient of performance of about 4.57 when producing cold water at a temperature of 1.1° C. In ice making mode, each chiller provides 1.5 MW_t at a coefficient of performance of 4.24 when producing cold water at a temperature of -4.4° C. In ice mode, the chiller capacity is reduced to 75% of its full capacity because of the low evaporative temperature.

The advantage of using an ice thermal storage system for this building is that this system has a heat recovery chiller that has a double bundle condenser. Using the ice storage as a heat sink, heat recovery supplies about two thirds of the building's annual heating energy. This is performed in a two-step sequence. First, the ice storage provides cooling for the interior zones in the building. As a result, the heat from the interior is transferred to the ice tanks. Second, in the evening, the heat recovery chiller removes heat from the ice tanks and sends it to the double bundle condensers heats the water to about 35.0° C and then pumps it to the AHUs. The use of ice thermal storage with a heat recovery chiller has eliminated the need for heat storage for heating, which would increase the initial costs, and has provided annual net electrical energy savings of 750 MW_eh, resulting in a simple payback of 2.0 years for the added cost of installing the heat recovery chiller.

The second and more important advantage of using ice thermal storage in this building is the colder chilled water temperature of 1.1° C, which is available from the ice storage system and which is used to produce the desired cold air to be distributed in the building. The possible colder water temperature available with ice storage reduces the temperature of the supply air at the AHUs to 6.7° C and reduces the volumetric flow rate of the supply air by 25% thus reducing the total fan power by 100 kW_e. As a result, the sizes and costs of the AHUs and ductwork are further reduced. Moreover, the use of cold air distribution has reduced the net cooling load by 106 kW_t and more than 1.1 MW_th per design day, and has improved comfort conditions by maintaining the relative humidity at about 40%.

3.8.1.2 Department store, Oxford Street, London

An external melt ice-on-coil storage system was installed for an expanded department store in Oxford Street, London (Crane, 1994). At the same time, the original cooling equipment was replaced by new screw compressor chillers. The expansion resulted in an increase in the cooling load from 1.6 MW_t to 3.0 MW_t, almost double the existing capacity. The ice storage system was designed to provide approximately 50% of the cooling requirement on a summer design day with a storage priority control strategy, so the increased cooling capacity was met without increasing the size of the chiller plant or the space required on the roof for cooling towers.

The cooling system of the store comprises three screw chillers, each rated at 603 kW_t when providing chilled water at 3.0° C in the day time mode (direct cooling). In the night time (charge) mode, each chiller provides 450 kW_t of cooling with a nominal chilled water temperature of –6.0° C. The ice storage system provides 9.3 MW_th of cooling during the discharge period. The ice/water is contained within a reinforced concrete tank that was constructed on the site. During installation, the concrete storage tank was lined to an inventory device and lined to the control system to measure the percentage of ice formed in the tank by measuring the change of water level in the tank.

The chillers were designed to start charging the storage tank over a 10 hour period starting at 9:00 hours. However, for days more similar to the design day conditions, the actual charging period is 9 hours because the water in the storage tank at the time of charging is close to its phase change condition. During the discharging cycle, the ice storage is operated to meet the cooling load as much as possible, and the remaining cooling requirement is met by the chiller. For days closer to the design day conditions with a peak load of 2.6 MW_t and total integrated load of 17.4 MW_th, the discharging cycle starts at 9:00 hours until 17:00 hours, the ice storage provides about 9.2 MW_th of base cooling load, and the chillers operate simultaneously to handle an additional of 8.2 MW_th. Furthermore, for a typical day operation, with a peak load of 1.8 kW_t and total cooling energy of 12.0 MW_th, the ice storage provides about 76% of the total cooling. Implementing an ice storage system in this site has shown a significant saving on the operating costs in comparison with providing all cooling on demand during the day due to the availability of a lower electricity tariff and more efficient cooling energy generation at night.

3.8.1.3 Dental clinic, Fort Bliss, Texas US

In a retrofit project, an ice harvester thermal energy storage system was installed in a dental clinic at Fort Bliss, Texas and came into operation in October 1990. The building had design cooling load ranges from 197 kW_t (from 12:00 hours to 13:00 hours) to 204 kW_t (from 15:00 hours to 16:00 hours). The ice harvester system consists of a 91.4 kW_t ice maker, 1.1 MW_th above ground steel storage tank (Sohn, 1991).

The clinic is a single story, rectangular building with a floor area of 5,639 m^2 approximately. It contains a dental services area, doctors and administrative offices, reception, and a supply storage facility. The cooling system consists of a 571.6 m^3s^{-1} air handler, a water cooled chiller nominally rated at 225 kW_t with two reciprocating 26.2 kW_e compressors, and a cooling tower.

The cooling system was designed to operate in such a way that from 20:00 hours to 5:00 hours the ice harvester chiller makes ice and all the AHUs shut down, so at this time period the building is not cooled. From 5:00 hours to 12:00 hours harvester is shut down, and the existing conventional water cooled chillers are switched on to directly cool the building. It is turned on two hours before the building is occupied. From 12:00 hours to 16:00 hours, the two chillers are switched off and the building cooling is met only by the stored ice. Finally, from 16:00 hours to 20:00 hours the chillers are switched on again and directly cool the building. The installation and successful operation of the ice harvester system achieve the primary purpose of shifting the electric power from the on-peak to the off-peak period to avoid the demand charge of 19.5 US$ per kW_e. However, the use of ice harvester system consumes about 29% more electrical energy than did the conventional cooling system.

3.8.2 Chilled water storage systems

3.8.2.1 <u>Electronic manufacturing facility, Dallas, Texas US</u>

A retrofit project was completed in August 1990 for the central chiller plant of a 106,092 m^2 electronic manufacturing facility in Dallas, Texas (Fiorino, 1991). In the project, a 10,161 m^3 stratified, precast, prestress, cylindrical concrete chilled water storage tank was installed to shift about 86.2 MW_th of cooling energy. The main motive for installing the water tank was to reduce the peak electrical demand of the facility by 2.9 MW_e, in order to cut annual electricity costs.

In the design of the chilled water storage tank, a double octagon diffuser was used instead of single octagon to increase the effective length of the diffuser system, hence reducing the inlet Reynolds and Froude numbers and therefore minimising the mixing of the stored water. Furthermore, flow splitters were added in the horizontal branch pipes at the mid point between the inner and outer octagons, and pipe reductions were taken at each flow splitter rather than at the inlets to the inner and outer octagons. This resulted in equal pressure drop characteristics of the diffusers, and a reduction of flow velocity and momentum by nearly 75% before the flow reaches the inlets to the inner and outer octagons. Moreover, the maximum flow velocity of 0.280 ms^{-1}, approximately equal to the recommended maximum outlet velocity of 0.274 ms^{-1} was maintained inside the linear diffuser pipes.

In this chilled water storage system, storage charging is performed at 20:00 hours until noon next day. All chillers and water pumps are operated in order to meet the facility's off peak cooling load, which ranges from 7.0 to 9.1 MW_t, and totals 122.4 MW_th cooling energy as well as charging the storage tank for the next day's on-peak cooling energy load, which totals 74.7 MW_th. During the charging cycle, the chillers are continuously running at full load, and the charging cycle is completed in about 13.4 hours. In order to keep the chillers operating at full load during the charging cycle, the temperature of the chilled water exiting the chillers was set to 3.3° C through the chillers' control panels, and the temperature of the chilled water exiting the externally throttling valves of each chiller was set to 4.2° C.

Parameter	Charging Cycle	Discharge Cycle
Duration (Minutes)	894.0	863.0
High Flow Rate (m^3s^{-1})	0.292	0.262
Low Flow Rate (m^3s^{-1})	0.200	0.168
Average Flow Rate (m^3s^{-1})	0.205	0.194
Total Flow Rate (m^3)	10,998	10,045
Tank Volume (%)	108.1	98.9
Start Inlet Temperature (°C)	10.3	14.4
End Inlet Temperature (°C)	3.4	18.0
Average Inlet Temperature (°C)	5.1	14.1
Start Outlet Temperature (°C)	14.1	5.6
End Outlet Temperature (°C)	5.6	14.0
Average Outlet Temperature (°C)	13.3	5.7
Average Temperature Difference (°C)	8.2	8.3
Average Energy Rate (MW_t)	7.0	6.8
Total Energy (MW_th)	104.9	97.2
Cycle Thermal Efficiency (%)	92.7	
Figure of Merit (%)	92.2	

Table 3.1 A summary of the thermal performance of the electronic manufacturing facility's central chiller plant.

During the discharging cycle, all chillers and their associated pumps are switched off at about 12:00 hours and the facility's cooling load is met only from the storage tank (i.e. full storage operation strategy). Constant speed transfer and variable speed distribution pumps are in operation to meet the facility's on-peak cooling load, which ranges from 8.9 to 9.8 MW_t. A summary of the thermal performance of the system is shown in Table 3.1.

For the same case study of the cooling system reported by (Fiorino, 1991), Fiorino (1994) demonstrated that the integration of the chilled water storage system has not only reduced the peak electricity demand but has also minimised the electricity and gas consumptions compared to when the system was running conventionally. Fiorino (1994) also reported that because of the correct design and the more efficient and improved integration of the chilled water storage tank in the cooling system, the facility has achieved the following objectives:

a. Shifting about 100% of peak cooling electrical demand to off-peak hours.

b. Reducing the peak electricity demand of the facility by 30.2%.

c. Reducing the cooling electricity annually by 28.3%.

3.8.2.2 Fort Jakson, South Carolina US

A chilled water storage tank of 8,517 m^3, capacity was installed for the central energy plant number 2 at Fort Jackson, South Carolina US. The system serves more than half of the Fort's cooling load (Chang, 1999; Chang, 1998). The purpose of incorporating the chilled water storage system was to shift the electrical demand of 3.0 MW_e, from the summer on-peak (i.e. 13:00 hours to 18:00 hours) to off-peak hours.

The central plant number 2 has four chillers, each rated at 4.2 MW_t, capacity with total flow rate of 0.524 m^3s^{-1}. The differential temperature of the chilled water is 6.7° C, with the temperature of the supply water of 5.5° C, and the temperature of the return water of 12.2° C. At night, the plant has a minimum cooling capacity of 7.0 MW_t to charge the tank.

The storage volume of 8,517 m^3 capacity was determined based on the total integrated cooling load of 590.9 MW_th (including 5% safety factor), a temperature difference between the supply and return water of 6.7° C, and an assumed figure of merit of 0.9 for the storage tank. An above ground vertical cylindrical shape concrete tank was chosen with a height of 13.4 m and a diameter of 29.9 m. A quadruple octagonal diffuser system with a total linear diffuser length of 259.4 m was chosen to ensure proper stratification. The upper and lower diffusers were identical in shape and both had an inlet height h_i equal to 0.1 m. A pressure drop of 137.9 kPa from the inlet flange to the outlet flange, and design flow rates of 0.252 m^3s^{-1} (charging) and 0.505 m^3s^{-1} (discharging) were specified.

Based on the temperature measurement of the chilled water inside the tank during the charging process, Chang (1998) found that the depth of the thermocline ranged from 0.305 m at the bottom level, to 0.457 m at the mid-level and 0.610 m at the top level in the tank, which were well within the recommended range of 0.3 to 1.02 m (Dorgan, 1994). Chang (1998) also calculated the theoretical efficiency of the charging cycle, and it was estimated to be 95%.

The use of a chilled water storage system in Fort Jackson central plant number 2 has reduced the peak electrical demand from 25.4 MW_e in 1995 to about 23.4 MW_e. For the first year of operation of the chilled water system, the system reduced the electrical cost for Fort Jackson from US$ 5.46 million in 1995-1996 to US$ 5.16 million in 1996-1997, and the total electrical bill was reduced by 0.3 million US$ during the first year of operation. The actual impact of the chilled water system was even more than that because during the 1996-1997 period, a number of large buildings were added to Fort Jackson (nine buildings with a total floor area of 31824 m^2. The actual saving excluding new buildings was estimated to be around US$ 0.43 million.

Other applications and case studies of incorporating ice and chilled water storage systems with AC systems of buildings are available in studies by (Seaman, 2000; Dincer, 2002b). Seaman (2000) provides several case studies of ice thermal storage systems for office buildings in London. The systems descriptions, seasonal performances, and operating costs are discussed in detail in the reference. Dincer (2002b) discusses several applications of ice and chilled water storage systems, heating thermal storage systems and cogeneration facilities.

3.9 Conclusions

In this chapter, the general advantages of cool thermal storage systems over conventional AC systems have been discussed. Also, this chapter has illustrated briefly the various types of cool thermal storage systems such as ice slurry, ice harvester, internal and external milt ice-on-coil, encapsulated ice, chilled water, eutectic salt storage systems and their various operation strategies.

An internal melt ice-on-coil system technology, which represents one of ice cool thermal storage systems, was chosen to be sized and examined for the clinic building. Chilled water storage is another storage technology that was selected to be studied its regarding its application to the clinic. The energy performances of the storage tank, the pumps interface, and the chiller in a chilled water storage system has been discussed in detail. It has been found that the diffuser in the chilled water storage tank is the most important element and must be carefully designed.

The chapter was concluded by reviewing several case studies of some existing ice and chilled water cool thermal storage systems. It has been found both systems can reduce the peak electrical demand as well as conserve energy if they are well implemented, designed, and operated in the cooling system.

In ice thermal storage systems, the use of a lower chilled water temperature for the production and distribution of the air in the building can have a great advantage. It can reduce both initial costs and electrical energy consumption of the cooling system.

It has also been found that chilled water storage systems can also reduce electrical cooling energy if the system is well designed and controlled. The electrical power consumption of the chilled water pumps can be minimised if the water distribution system is well maintained at a high temperature differential. Furthermore, the performance of the chilled water storage can be improved by careful design of the diffuser system, and good insulation of the storage water tank.

From the existing case studies discussed in this chapter, it was found that ice and chilled water storage systems reduced the electrical energy consumptions during peak power demand time, therefore, reducing the operating costs. Ice storage systems were found to be used with small and medium size buildings. In ice storage systems, air distribution system were designed with colder supply air by supplying colder chilled water temperature of about 1.0° C to the AHUs. With colder supply air, the volumetric air supply was reduced thus reducing the total fan power and connected power. However, in some existing building (e.g. See Section 3.8.1.3), the use of ice storage system increased the energy consumption by 29% relative to conventional system.

Moreover, it has been found from the existing case studies for chilled water storage systems that these systems were ideal for increasing the capacity of existing conventional systems since it uses standard chillers with no need for special equipment. They are suitable for typical large buildings where large electrical energy is shifted from day to night time. Furthermore, chilled water storage systems were found to have lower electrical energy consumption when compared with conventional system (e.g. See Electronic manufacturing facility).

Chilled water storage systems are always associated with large water tanks; within the storage tanks the diffusers are built. The diffusers are the most important element of the chilled water tank because the performance of the storage tanks depends on their design. Tanks with good diffusers design can perform with a figure of merit of more than 90% (see Table 3.1). From the existing chilled water storage system valuable lessons were learned specifically regarding tank sizing, tank configuration and internal diffuser design and can be used to design the diffusers of the storage tank for the clinic.

Chapter 4

Energy Simulation of the Clinic

4.1 Introduction

The main aim of this chapter is to develop hourly cooling load profile for the clinic building for the design day and whole year. This is required to size the various components in the conventional, ice and chilled water storage AC systems and therefore to estimate the energy consumption of each system. Furthermore, using the hourly load profile will help to determine the occurrence of maximum cooling load during the day time, hence selecting appropriate charging and discharging hours for the requirements of chiller and storage sizing.

The first step in sizing an AC system for the clinic building is to develop an hourly cooling load profile. The ESP-r building energy simulation program (Clarke, 2001) is one of the available tools that can be used to obtain the profile. This chapter discusses the input data that is required to simulate the building. The data include building geometry, construction, orientation, ventilation, and boundary conditions as well as the number of occupants, lights, and appliances and their operation schedules.

The simulation was carried out in at the simulation lab in KISR. The results of the simulation were analysed for both controlled and uncontrolled ventilation and a suitable load profiles was developed for the clinic so that the AC systems can be sized. Other cooling load profiles were also obtained and examined for other months in the year; this will help later to estimate the energy consumption of the various cooling systems under study for the whole year. The maximum and integrated cooling load obtained from the simulation was validated with the actual load obtained for the clinic.

4.2 Introduction to building energy simulation ESP-r

Computer based simulation is one of the most powerful methods available for determining hourly cooling loads for a building. A computer model can be created for the building using existing plant drawings to define the vertices and the surfaces of the spaces within it. The surface within a building can include internal and external walls, roofs, ceilings, and floors and so on. The type of materials used in the surfaces can be specified and their thermo-physical properties can be defined. Furthermore, based on the zone operating hours, lighting, occupancy, and equipment load profiles can be defined for each zone within the building. The model is then subjected to some boundary conditions such as weather data and interior temperature and humidity settings of the air conditioned space to estimate the hourly cooling load for the building for the whole year.

One of the most powerful building simulation codes is ESP-r (Clarke, 2001). This state of the art, whole building simulation software, is used by the European Commission as a reference program for building energy simulation. The software is based on the integrated dynamics simulation in which the thermal performance of the building is systematic, that is, different heat transfer mechanisms such as conduction, convection, and radiation interact in a complex manner within the building. The use of such an advanced building simulation tool for predicting the cooling load is necessary because the thermal analysis of a building is complex and must be calculated and analysed accurately. Accurate prediction of the cooling load is a very important task since the selection, sizing, and successful operation of cooling systems with or without cool thermal storage depends on the accuracy of estimating the building cooling load profile. Furthermore, if the cooling load is over estimated, this will result in excessive cooling system capacity and thus higher capital investment and demand for power and energy.

4.2.1 General purpose of ESP-r

ESP-r is transient energy simulation system software that is capable of modelling the energy and fluid flows within combined building and plant systems. The software contains a number of interrelating program modules addressing project management, simulation, results recovery and display, database management and report writing.

One or more zones within the building can be defined in terms of geometry, construction, and usage profiles. These zones are then combined to form a building, either entirely or in part, and the leakage distribution is defined optionally to enable air flow simulation. The plant network is then defined by connecting individual components.

And, finally, the single or multiple zones building and components are connected and subjected to simulation processing against user defined control. The entire data preparation exercise is achieved interactively and with the help of pre-existing databases that hold the standard or user defined constructions, event profiles of the building, and its plant components.

ESP-r can be used to simulate existing and new buildings; it offers input or output facilities that give detailed answers to the users for design questions such as

a. When do the building or plant loads occur?
b. What will be the effect of a certain design change, such as wall insulation, window shape and size, type of glazing, re-zoning of the building, or change in the heating/cooling control regime?
c. What will be the comfort levels within the building?
d. How do suggested design alterations affect air flow and fresh air distribution within the building?

Other questions related to the effect of ventilation and infiltration loads on the system load and controls of heating and AC systems can be answered.

In common with other simulation programs, the ESP-r approach is markedly different from traditional methods in that it aims to represent all relevant phenomena, and to process these phenomena simultaneously so that the inter-relationships are preserved. Essentially, this is achieved by establishing sets of conservation equations for different spatial regions and arranging for the integration of these equations over time. In this way, the energy and mass flows are tracked throughout a simulation as they change under the influence of climatic boundary conditions, occupancy effects inside the building, constraints imposed by any control action and by the potentially time-dependent inter-nodes links (representing, for example, damper, valve, and window or door movement).

The theories employed by ESP-r to represent heat transfer and fluid flow, and the numerical techniques used to achieve equation integration, are given in detail in the study by (Clarke, 2001).

4.2.2 Basic structure of ESP-r

The relationship between the program and the database module that form the ESP-r simulation environment is shown in Figure 4.1., ESP-r comprises three principal modules: a Project Manager helps the user to define the building and plant configuration and controls user access to the other ESP-r programs such as Simulator and Results Analyser; a Simulator models the problem containing building, plant, and control components, and a Results Analyser, which analyses the time series of state variables as generated by the Simulator.

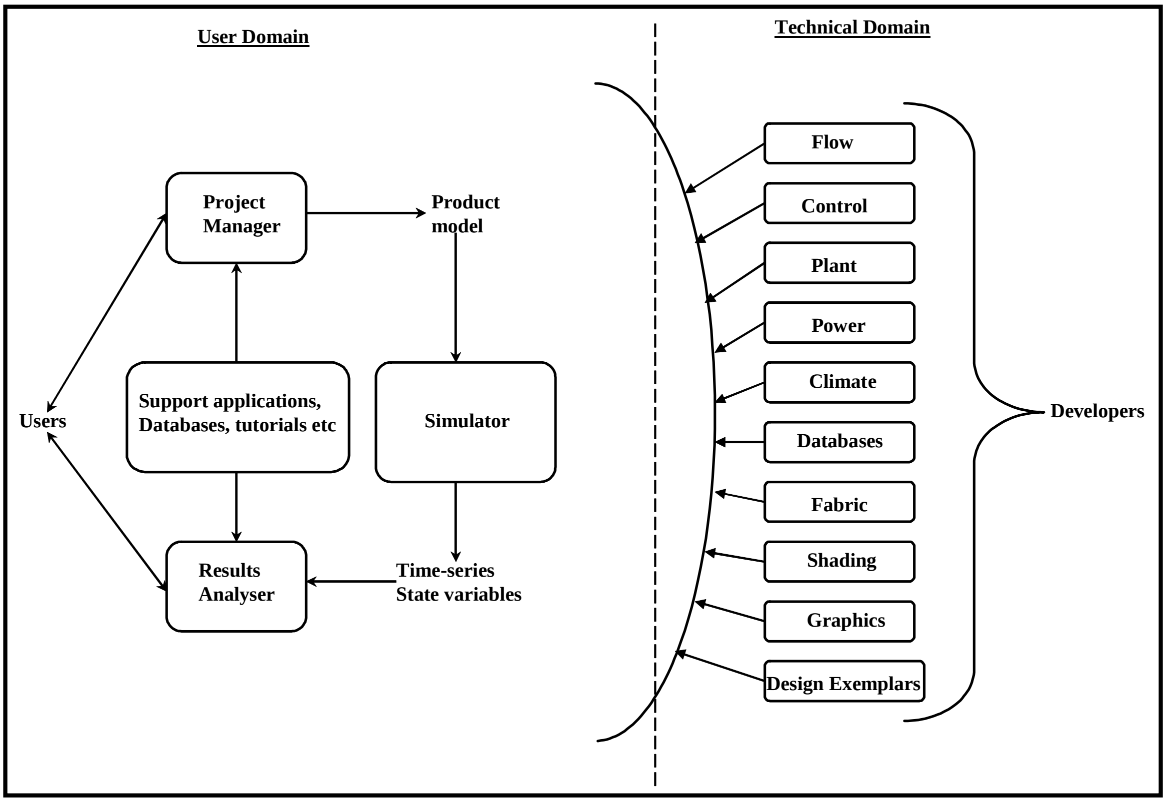

Figure 4.1 ESP-r structure.

4.3 ESP-r simulation input data

The energy simulation program ESP-r was used to predict the cooling load of the clinic. The first task in the simulation was creating and defining the materials, multilayer, event profiles and climate databases, and then a computer model was created using existing plant drawings. In this model the vertices were defined for each surface within the building and surfaces were created that represented all the walls in the building envelope including the windows. The geometry, attributes and construction were defined for all surfaces using existing databases within ESP-r. The proper boundary conditions including the variation in soil temperature, dry bulb temperature, direct normal radiation, diffuse radiation, wind speed, wind direction and relative humidity were applied at each boundary surface.

4.3.1 Using the databases

A number of standard databases are available within ESP-r. For some simulations, a user may have to define his or her own databases. In this case, the user may copy, use available ESP-r standard databases and apply the required modification, or simply start from scratch. For the clinic, the standard databases of ESP-r including materials, multilayer constructions, climate and event profiles databases were used and modified according to the available data of the clinic.

4.3.1.1 Material types and their thermophysical properties

In the material database, the definition and description of the thermophysical properties of the materials such as stone, insulation, windows and so on that were used in the clinic were built up. The thermophysical properties, including density, specific heat, conductivity, and diffusion resistance, surface absorbivity, and emissivity, were defined for each constructed material.

4.3.1.2 Multilayer construction

In the multilayer construction database, the composition of the construction for the walls, floor, ceiling, ground floor, roof, and windows in terms of their constructed layers of materials of particular thickness in a particular order, with a named set of optical properties were defined. Each item (that is, constructed layers) in the database was given a unique name, accessed, and assigned to a surface by its name. For an asymmetrical construction including the internal walls, partitions, and ceilings that were located between two thermal zones, an inverted version of the construction was created with a slightly different name. This action was necessary because the ceiling construction must be built up from the outer surface to the inner surface that is facing the zone. Table 4.1 summarises the properties of construction materials that were used in the clinic.

Wall Type	Layer	Thermal Conductivity ($Wm^{-1}K^{-1}$)	Density (kgm^{-3})	Specific Heat ($Jkg^{-1}K^{-1}$)	Thickness (m)
Roof	Mosaic tile	1.104	2284	795	0.020
	Cement plaster	1.000	2085	837	0.020
	Sand cement screed	1.000	2080	840	0.030
	External polyurethane	0.029	46	1214	0.020
	Water proofing	0.140	934	1507	0.003
	Sand cement screed	1.000	2080	840	0.020
	Foam concrete	0.210	351	879	0.050
	Concrete slab	1.770	2297	921	0.150
External Wall	Sand lime brick	1.310	1918	795	0.110
	Cement block	1.640	2011	921	0.100
	External polyurethane	0.032	30	1214	0.019
	Cement block	1.640	2011	921	0.150
	Cement plaster	1.000	2085	837	0.020
Floor	Concrete slab	1.770	2297	921	0.150
	External polyurethane	0.031	46	1590	0.070
	Kuwait sand	0.340	1800	800	0.060
	Sand cement screed	1.000	2080	840	0.020
	Mosaic tile	1.104	2284	795	0.020
Ceiling	Mosaic tile	1.104	2284	795	0.020
	Sand cement screed	1.000	2080	840	0.020
	Kuwait sand	0.340	1800	800	0.060
	External polyurethane	0.031	46	1214	0.070
	Concrete slab	1.770	2297	921	0.150
Internal Wall	Cement plaster	1.000	2085	837	0.020
	Cement block	1.640	2011	921	0.150
	Cement plaster	1.000	2085	837	0.020

Table 4.1 Thermophysical Properties of Construction Materials

4.3.1.3 ***Event profiles***

Distinct named profiles describing the time dependent behaviour or zone internal heat gain processes from lighting, appliances, and people within a twenty-four hour period were defined in an event profile database. For each zone in the clinic, a separate profile was defined for people, lighting and appliances. The occupancy profile was developed based on the percentage of total attendances in the clinic which was based on the expected number of attendances entering and leaving the building for each hour. Similarly, the profiles of the lighting and appliances were estimated based on the expected number of lights and PCs switch on.

For example, the occupancy profile was defined in ESP-r for each zone by specifying the percentage relative to their total occupancy attended at each zone and at each hour of the day. For example, from 0:00 hours to 5:00 hours and from 15:00 hours to 11:00 hours the zone is unoccupied; this means 0% of the total occupancy is present in the zone; from 7:00 hours to 8:00 hours and at 14:00 hours, 70% of the total attended people are occupied in the zone; and finally from 9:00 hours to 13:00 hours, 80%. The profiles for lighting and appliances were defined in the same manner; however, their percentages were specified in terms of their total number and connected power in the zone. Figure 4.2 shows an example for the occupancy, lighting, and appliances profiles for zone number one in the building.

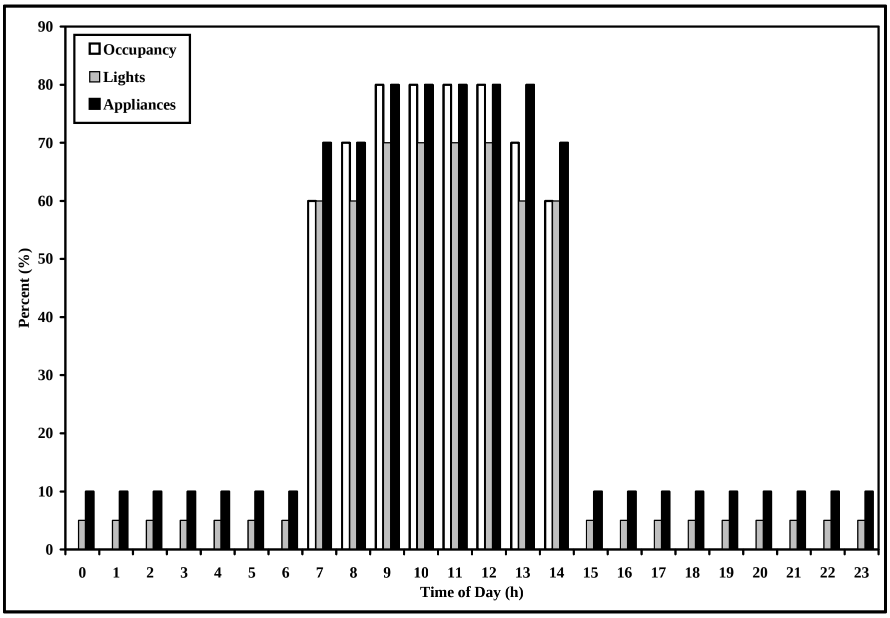

Figure 4.2 Occupancy, lighting, and appliances profiles of zone number one.

Furthermore, for zone number one, separate profiles were assigned for weekend days; it was assumed the building is unoccupied at the weekend, 5% of the total number of lights is on, and 10% of the total number of appliances is on. All other zones were having the same occupancy, lighting, and appliances profiles as for zone number one for both week and weekend days, but with different total numbers of people, lighting, and appliances.

4.3.1.4 Climate

Hourly sets of values from 1 hour on 1 January to 23 on 31 December were defined in the climate databases. For each hour, the following data was held:

a. Diffuse horizontal solar intensity (Wm^{-2})

b. Dry bulb temperature (tenths of degree C i.e. 102 = 10.2° C)

c. Direct normal or global horizontal solar intensity (Wm^{-2})

d. Wind speed (tenths ms^{-1} i.e. 15 = 1.5 ms^{-1})

e. Wind direction (degrees from north, clockwise)

f. Relative humidity (%)

4.3.2 Building coordinates and dimensions

Before inputting the data to account for the internal and external heat gain within the clinic building, the building geometry was built. The zones in the building were constructed separately and each was described as an extruded shape and supplied with the floor and ceiling heights as well as the coordinates of the various corners and connections between these corners. The coordinates and dimensions for each zone and for the whole building were obtained by using existing drawing plants. Figure 4.3 shows the view of the whole clinic building in ESP-r, and the location of each zone after the coordinates were defined, and the surfaces were created.

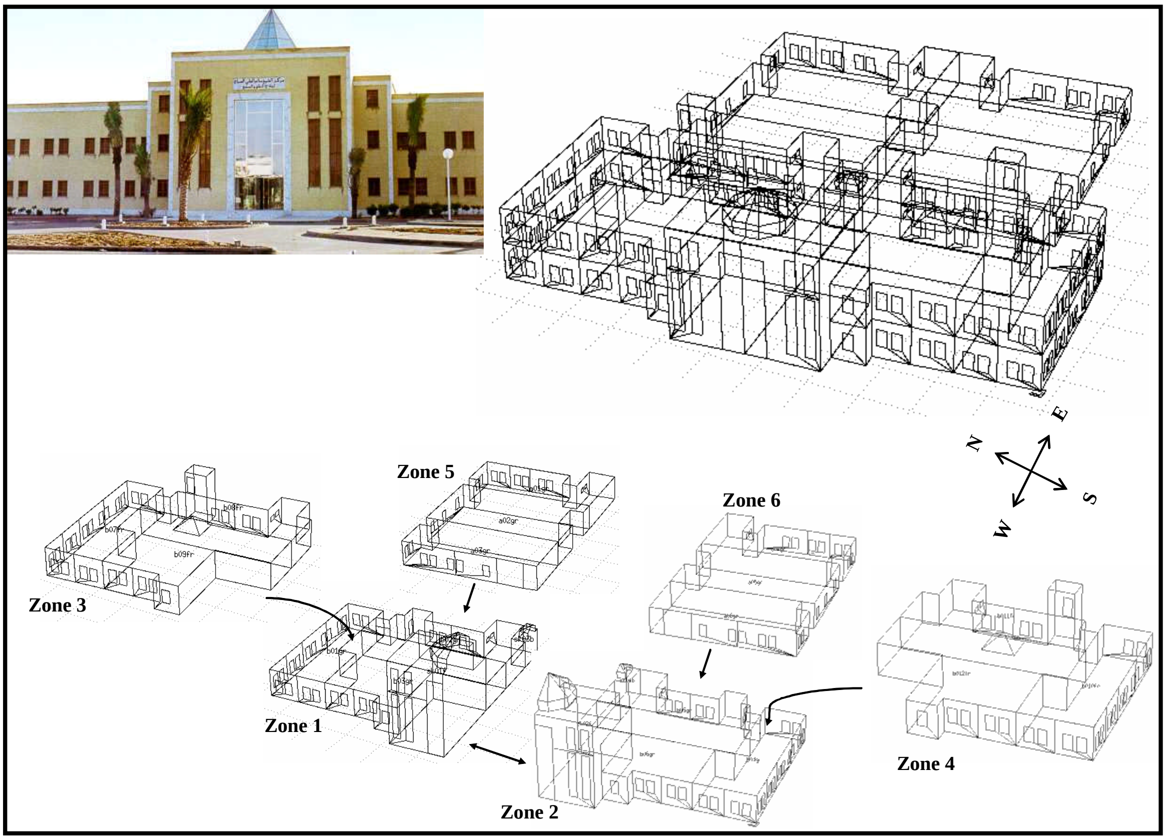

Figure 4.3 Geometry of the clinic building in ESP-r.

4.3.3 Internal heat gain

4.3.3.1 Occupancy

The total occupancy for each zone is given in Table 4.2. The heat gain from occupancy has two components: sensible and latent. The amounts of sensible and latent heat vary with the level of activity within the building. Generally, latent heat gain increases as the level of activity increases. For the clinic, it was assumed that people’s activity level is seated light work. The adjusted heat gain was based on a normal percentage of men, women, and children and was taken from ASHRAE Handbook (ASHRAE, 2001c).

Zone Number	Number of People	Occupancy Heat Gain (kW_t)	Sensible (kW_t)	Latent (kW_t)
Zone 1	79	9.1	5.6	3.6
Zone 2	78	9.0	5.5	3.5
Zone 3	64	7.4	4.5	2.9
Zone 4	67	7.7	4.7	3.0
Zone 5	85	9.8	6.0	3.8
Zone 6	72	8.3	5.1	3.2
Total	445	51.4	31.4	20.0

Table 4.2 Total occupancy and the heat gain for each zone in the clinic

The heat gain from an adult male is 130.0 W_t, and that from the adult female and from children is 85% and 75% respectively that of the adult male (ASHRAE, 2001c. Therefore, the average heat gain which accounts for the male, female, and children is 115.0 W_t. Note that the heat gain is rounded to the nearest 5.0 W_t. The overall heat gain due to the occupancy was estimated for each zone by multiplying the average heat gain of 115.0 W_t by the total number of occupancy in zone. Since the sensible heat gain is 61% and latent heat gain is 39% of the total heat gain (ASHRAE, 2001c), therefore, the sensible and latent heat gains were respectively equal to 70.5 W_t and 45.0 W_t of the average heat gain.

Other important information that was defined in ESP-r was the radiation and convection parts of the sensible heat gain from the occupancy; it was assumed that the convective part was 30% of the total sensible heat gain and the radiative was 70% (McQuiston, 1992).

4.3.3.2 Lighting system

The total lighting wattage for each zone in the clinic is given in Table 4.3. Lighting is a major internal load component. The source of the heat from the lighting comes not only from the light emitting element, or lamps, but also from its associated components such as chokes and fixtures.

Zone Number	Power (W_e)
Zone 1	7197
Zone 2	7792
Zone 3	7771
Zone 4	8291
Zone 5	5822
Zone 6	4778
Total	41649

Table 4.3 Total lighting connected wattage for each zone

Most of the lighting in the clinic building was of the fluorescent type. Fluorescent lights convert about 25% of the power input into light, with about 25% being dissipated by radiation to the surrounding surfaces. The other 50% is dissipated by conduction and convection (Carrier, 1965).

The fluorescent fixtures installed in the clinic have four fluorescent tubes of 18 W_e each and two chokes of 2 W_e each. Thus, each fixture has a total wattage of 76 W_e. Other fixtures have two tubes of 36 W_e each and two chokes of 2 W_e each. The total wattage of these types of fixtures is also 76 W_e.

The electricity consumed by the chokes of the fluorescent light converts into heat and adds to the building load instantly. The electricity consumed by the tube, however, transforms into radiation (delayed portion) and convection (instant portion). McQuiston (1992) assumed that the heat gain to the conditioned space from fluorescent fixtures is 59% radiative and 41% convective. The input data in ESP-r of the lighting system in the clinic was, therefore, the total electrical wattage consumption of the lights in each zone of the clinic and their associated percentages of radiation and convection.

4.3.3.3 Appliances

In the estimation of the cooling load of a building, the heat gain from all appliances within the building should be taken into account. The estimation of the heat gain from the appliances is very subjective. This is because of the variety of their applications, schedules, use, and installations. For most of the appliances, the only information available about heat gain from the equipment is given on its nameplate.

Offices with computers, monitors, printers, check writers and other heat generating equipment can generate 9 Wm^{-2} to 13 Wm^{-2} (McQuiston, 1992). Many recommended heat gains from typical computers, laser printers, copiers, and other miscellaneous office equipment such as folding machines, inserting machines, coffee makers and so on are available in a tabulated form and given by (ASHRAE, 2001c). The heat gain of the office equipment in these tables was obtained based on a generalised analysis of the actual measurements from different individual researches and ASHRAE projects. However, the results that were obtained for the laboratory and hospital equipment was found to be too diverse and could not be generalised; however, heat gain for most of the hospital and laboratory equipment is available in the literature (McQuiston, 1992; ASHRAE, 2001c)

A proposed study for the estimation of heat generation from different equipment was discussed by (Dunn, 2005) and was based on the occupancy density. His results showed that the heat gain from small power equipment ranges between 6 Wm^{-2} and 34 Wm^{-2} per treated floor area and ranged between 124 Wm^{-2} and 229W_t per person when it is normalised for occupancy.

The Heating Ventilation and Air Conditioning (HVAC) designer of the clinic (BLK, 2000) assigned a total number of 30 personal computers (PCs) (computer and monitor) in the clinic, each with an assumed heat gain of 300 W_t. However, the connected power for other appliances was not available in the report. Because of the difficulty of accessing the clinic due to the presence of highly sensitive and expensive testing devices that are used to test patients' abilities in speech, hearing, and observation, the total numbers of PCs in the simulation was assumed to be 45 instead of 30 as reported in British Link Kuwait (2000) to account for other appliances that were included in the report. The total number of PCs and their assumed heat gain for each zone are given Table 4.4.

Zone Number	Number of PCs	Power (W_t)
Zone 1	6	810
Zone 2	10	1350
Zone 3	11	1485
Zone 4	4	540
Zone 5	7	945
Zone 6	7	945
Total	45	6075

Table 4.4 Number of PCs and their heat gain in each zone

Furthermore, a conservative heat gain of 65 W_t was selected for the computer, and medium size monitor (40 mm to 640 mm) of 70 W_t. Thus, the total heat gain from each PC was 135 W_t (ASHRAE, 2001c), which is less than the value given by the HVAC designer (BLK, 2000). The PCs have only sensible heat gain: the radiation part is between 20% and 30%, and convective part is between 70% and 80%; by taking the average values, the radiation was assumed to be 25% and the convection to be 75%.

4.3.4 Infiltration and ventilation

An uncontrolled flow of air through small openings such as cracks in walls and ceilings, and through the perimeter gaps of windows and doors is known as infiltration. This air flow is driven by wind, the temperature difference between inside and outside the building, and internally induced pressure within the building. Infiltration through the building increases the building cooling load because of the high ambient outside dry bulb temperature and relative humidity that might enter the building or zone. Most of the HVAC designers in Kuwait over pressurise the air systems in buildings, which means that air is usually leaving the building (exfiltration). This means the amount of fresh air that enters the air handling units is slightly greater than the amount of exhaust air exiting the AHUs. For that reason, it was assumed that the infiltration through the building was zero.

Ventilation is the filtered outdoor air flow that is introduced through the cooling system by mechanical fan, and is mixed with the conditioned return air, and then supplied to the space by the air handling units. ASHRAE (1989) recommends a minimum flow of

0.008 m^3s^{-1} per person of uncontaminated outside air to be supplied in an occupied space, and in bathrooms of 0.03 m^3s^{-1}. Table 4.5 shows the ventilation rate in m^3s^{-1} for each zone based on the total number of people and toilets.

Zone Number	Number of People	Ventilation (People)	Ventilation (Bathroom)	Total Ventilation
Zone 1	79	0.57	0.008	0.65
Zone 2	78	0.56	0.008	0.64
Zone 3	64	0.46	0.008	0.53
Zone 4	67	0.48	0.008	0.56
Zone 5	85	0.62	0.005	0.67
Zone 6	72	0.53	0.005	0.58
Total	445	3.22	0.400	3.62

Table 4.5 Recommended ventilation rate in m^3s^{-1} for each zone

4.3.5 *Boundary conditions*

For each surface within the building, a proper boundary condition was defined. The surfaces have two sides, one facing the zone (inside) and the other connected to a boundary condition (another zone, ground, outside). They interact both radiantly and convectively with their environment. A surface may be opaque or transparent.

The boundary condition was defined as the ground floor surface of the building using a monthly temperature profile that represents the variation in the soil temperature in Celsius as shown in Table 4.6. The temperature values were obtained from the Annual Climatological reports and they represent the average for 1970, 1972, 1985, and 1986.

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
19.5	19.2	20.7	24.6	28.3	32.4	34.4	35.7	34.8	32.3	28.2	22.5

Table 4.6 Monthly ground average temperature in °C

In order to simulate the energy performance of the clinic, the energy simulation program ESP-r requires a set of one year (that is, 8760 hours) of weather conditions. The typical meteorological year that was developed for Kuwait conditions (Shaban, 2000) was saved in the climate database, and was used during the simulation.

The typical meteorological year represents a yearly weather profile and was generated based on categorising twelve typical meteorological months of the real time meteorological data. It contains an hourly meteorological weather data for the whole year such as dry bulb temperature, direct normal radiation, diffuse radiation, wind direction, wind speed and relative humidity. These were the external time dependent boundary conditions and were used to define the conditions at the external wall surfaces, roof, and windows.

Other important information that was required for the simulation was the latitude and longitude of the site of the clinic. These were defined and were assumed to be similar to the city of Kuwait, (that is, 29.3° latitude and 47.9° longitude). Finally, to continue with the inputting of the boundary conditions, the simulation period was set to be from Thursday 1 of January until Friday 31 of December. An ideal controller with a maximum heating capacity of 999999 W_t and minimum of 0 W_t, and maximum cooling capacity of 999999 W_t and minimum of 0 W_t were applied. Furthermore, the heating and cooling temperature settings were 21°C and 24°C respectively. If the building cooling or heating loads had exceeded the maximum value of the cooling and heating capacities defined above, the maximum cooling capacity would have had to be increased and the building would have had to be re-simulated.

4.4 Simulation results and analysis

The simulation was performed via the simulator within the ESP-r program. Many results were obtained and analysed including hourly occupancy, lighting and appliances load profiles, and more importantly, the hourly cooling load profiles for each zone and the whole building for the peak month and the whole year.

4.4.1 *Internal load profiles*

4.4.1.1 Occupancy

Occupiers in the building generate heat due to metabolic processes. The released heat from their body surfaces to the zone or building is by radiation, convection, and evaporation. The amount of heat released depends on the activity level of the people and the surrounding temperature. A high activity level and a low surrounding temperature, result in more heat generation.

Generally, it is very difficult to estimate the total number of people in the building for each hour in the day because the number of people varies from day to day and from hour to hour. It was assumed that the number of patients was about 80% of the total expected patients and members of staff were occupying the building between 9 hour and 12 hour, 70% at 13:00 hours and 60% at 14:00 hours. The shape of the load profile as shown in Figure 4.4 was generated based on the assumed occupancy profile (see Figure 4.2). Other profiles to describe the time dependent behaviour of people in the building are available in ESP-r. Figure 4.4 shows the amount of sensible and latent heat released by people at different hours in the day. The sensible heat gain represents about 61% and the latent heat gain represents about 39% of the total heat gain.

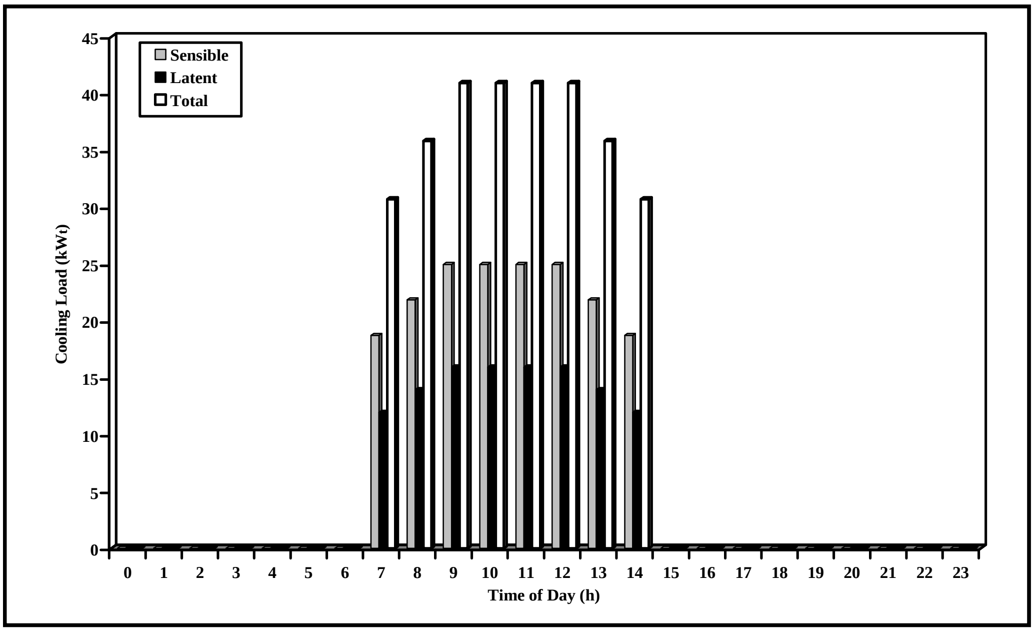

Figure 4.4 Occupancy load profile in the clinic.

Figure 4.4 shows the hourly cooling load resulting from the heat gain by the occupancy in the building for a 7-hour period from 7:00 hours to 14:00 hours. The load profile is, in fact, similar to the occupancy schedule that has already been defined in the event profile database. The maximum load from the occupiers depends on the total number of people present in the building and on the assumed diversity factor. The total number of people in the building was 445, and the heat generation per person was 115.0 W_t, so the total heat generation from the occupancy was 51.4 kW_t. Assuming diversity of 80%, which means that not all the people are present in the building at the same time, but only 80% of them are present, the load is equal to the multiplication of 51.4 kW_t by 0.8, which is about 41.1 kW_t as shown in Figure 4.4 between 9:00 hours and 12:00 hours. The heat gain by the occupancy for other hours in the day can be calculated in the same manner. For example, at 14:00 hours, it was assumed that only 60% of the total number of people is present, thus, the load is therefore equal to 30.8 kW_t (e.g. 51.4×0.6).

4.4.1.2 Lighting

Lighting is also a major source of heat in the building. It converts the electrical power input to light and heat. Unlike occupancy, lights generate only sensible heat. This heat is dissipated to the surroundings by convection and radiation and by conduction through the lighting fixtures to the adjacent surfaces such as the ceiling onto which the lights are usually fixed.

The heat gain from the lighting system is shown in Figure 4.5. The lighting load increases during the occupancy period from 7:00 hours to 14:00 hours. During the non-occupancy period, only 5% of the lights are on representing the emergency lights and other lights that are left on by some people. The maximum heat gain by the lights that occurs at the peak load time is about 29.1 kW_t. This value is calculated based on a diversity factor of 70%, and results in a calculated total heat gain from the lighting system of 41.6 kW_t. The diversity is applied here for the lighting system since it is not expected that all the lights are kept on during the occupancy period, because some areas such as the reception, the waiting room, and some of the corridors have natural light from the skylight windows.

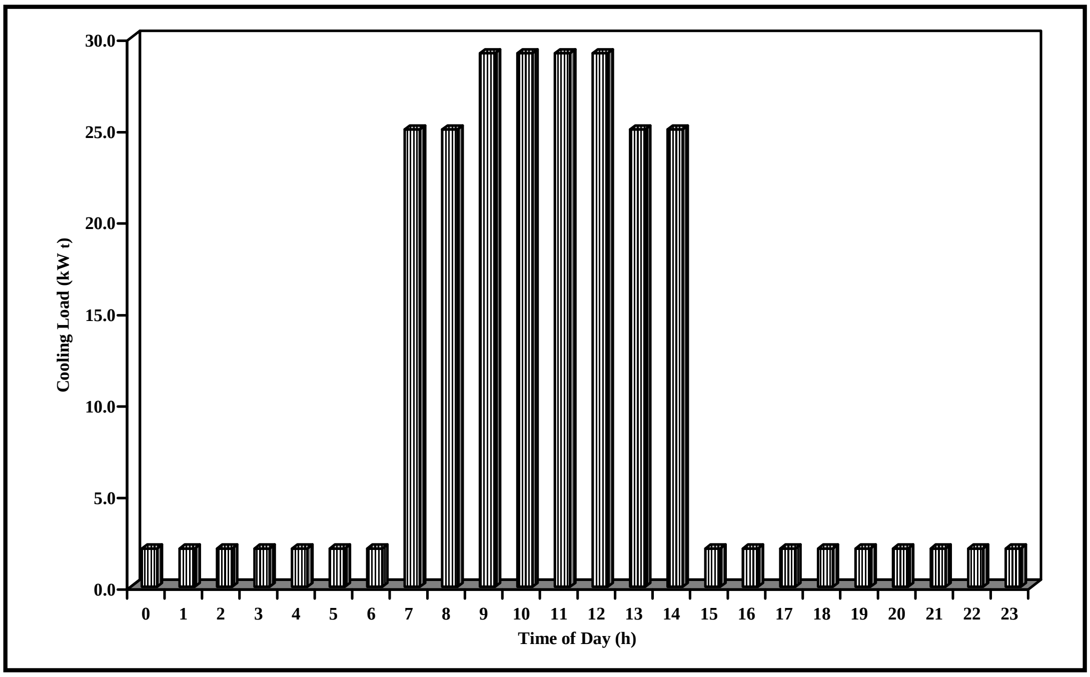

Figure 4.5 Lighting load profile in the clinic.

4.4.1.3 Appliances

As mentioned in section 4.4.4 of this chapter, the PCs are the only appliances considered in the calculation of the cooling load; each generates about 135 W_t of heat and there is no drying, cooking, or laboratory equipment that can generate water vapour present in the building, therefore, all the heat gains are of the sensible type. The total load of the PCs is 6.1 kW_t; assuming only 80% of the PCs are on, the load at the peak hour time is 4.9 kW_t, as shown in Figure 4.6.

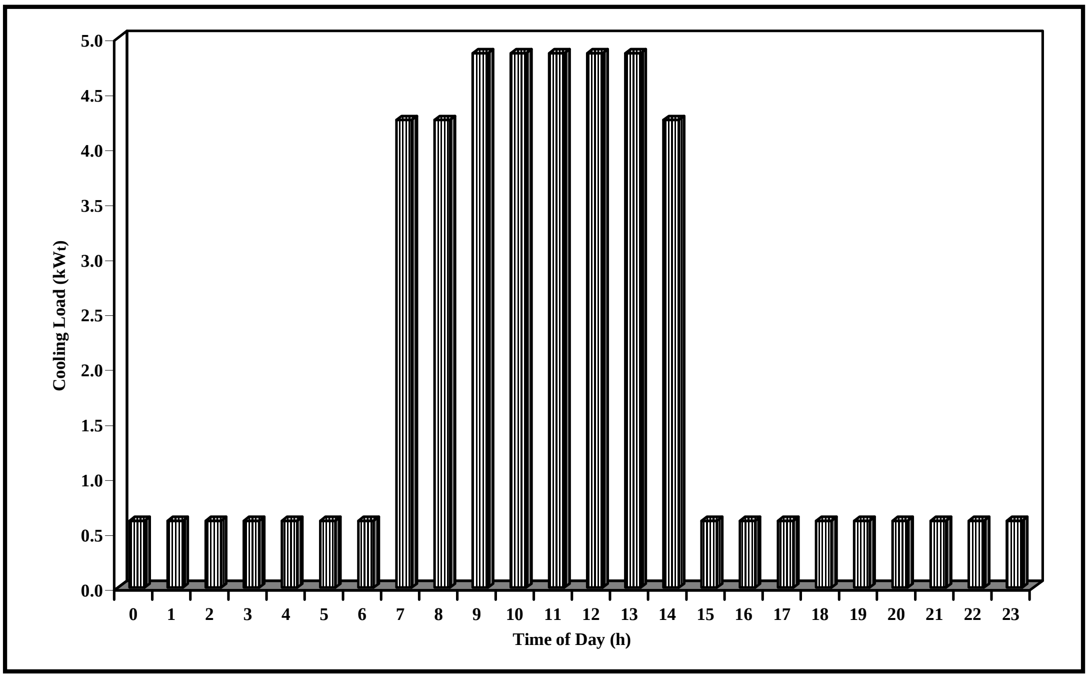

Figure 4.6 Appliances load profile in the clinic.

Furthermore, Figure 4.6 shows that during the non-occupancy period, the lighting load is small (about 0.6 $kW_{t)}$, as it was assumed in the definition of the appliances load profile that only 10% of the PCs would be on during the non-occupancy period since some people leave their PCs on and do not turn them off while they are not in their office. In fact, the load from the appliances has the lowest load compared with other loads such as occupancy, lights, fresh air, and so on. However, in some cases, appliances make a large contribution to the building cooling load. These cases can be found in buildings with a very large number of computers such as computer centres, and with many large pieces of laboratory equipment, such as hospitals.

4.4.2 Zones and whole building load profiles

The main purpose of simulation is to obtain the cooling load profile for the zones and therefore for a whole building; this helps to size the different components of the AC systems including AHUs, pumps, and chillers, and so on. Sizing each component will be discussed in detail later in Chapter 5.

Using the ESP-r building simulation program, a full year simulation was conducted for the whole building with controlled and uncontrolled ventilations. Several cooling load profiles were examined for different days in the peak summer months of June, July, August, and September to obtain a suitable load profile that could be used to size the components for conventional and cool thermal storage AC systems. It was found that the load profiles for some of the days were not uniform. So it was quite difficult to obtain a common load profile that could be used to size for both the AC systems with and without the cool thermal storage because in a conventional AC system, the peak cooling load that usually occurs at any peak hours (for example, between 12:00 hours to 16:00 hours) is the only information required to size the system. However, in the case of cool thermal storage, the size of the AC systems depends on the total integrated cooling load of the building. For that reason, using the load profile of a single day in the summer months obtained from ESP-r that has the maximum cooling load cannot be used to size both AC systems because it cannot be guaranteed that the day on which the maximum load occurs is also has the maximum daily total integrated cooling load when it is compared to other days in the year, especially in the peak summer days.

It is strongly recommended here to develop a suitable single profile that can be used to size both systems in order to make a reasonable comparison between their energy performances and their initial and operating costs. Thus, the average hourly cooling load of the building, excluding the ventilation load, was taken for each month in the year; the ventilation load was not required here because the weather data in ESP-r does not represent the outside weather design conditions at which the ventilation load should be calculated to determine the load on the cooling coil of the AHUs. However, the ventilation load that was obtained from ESP-r will be used later to analyse the energy performance for the design of different AC systems later in Chapter 6.

It was found that the hourly average cooling load in the month of August had the maximum cooling load of 208.5 kW_t, and the maximum total integrated cooling load of 3.4 MW_th. When the ventilation load was included in the analysis, the results showed that June had the maximum cooling load of about 329.4 kW_t, and was greater than August by 0.5%; the total integrated load was 5.3 MW_th and thus was lower by 6.5% compared to August.

4.4.2.1 *Zone profiles*

Since the building cooling load (without ventilation load) in August had highest both maximum and total integrated cooling loads, it was selected for sizing the components of the AC systems in the clinic. Figure 4.7 shows a plot of the average load profile for each zone in the clinic without ventilation versus time of day. The figure shows that zones 2 and 3 have the lowest maximum loads at the peak time, since the upper and lower boundaries of these zones are not subjected to solar radiation and the high ambient conditions (see Figure 4.3 for the zones' locations). The upper boundaries are facing the ceiling (that is, the ground of the first floor), and the lower boundaries are facing the ground floor of the building.

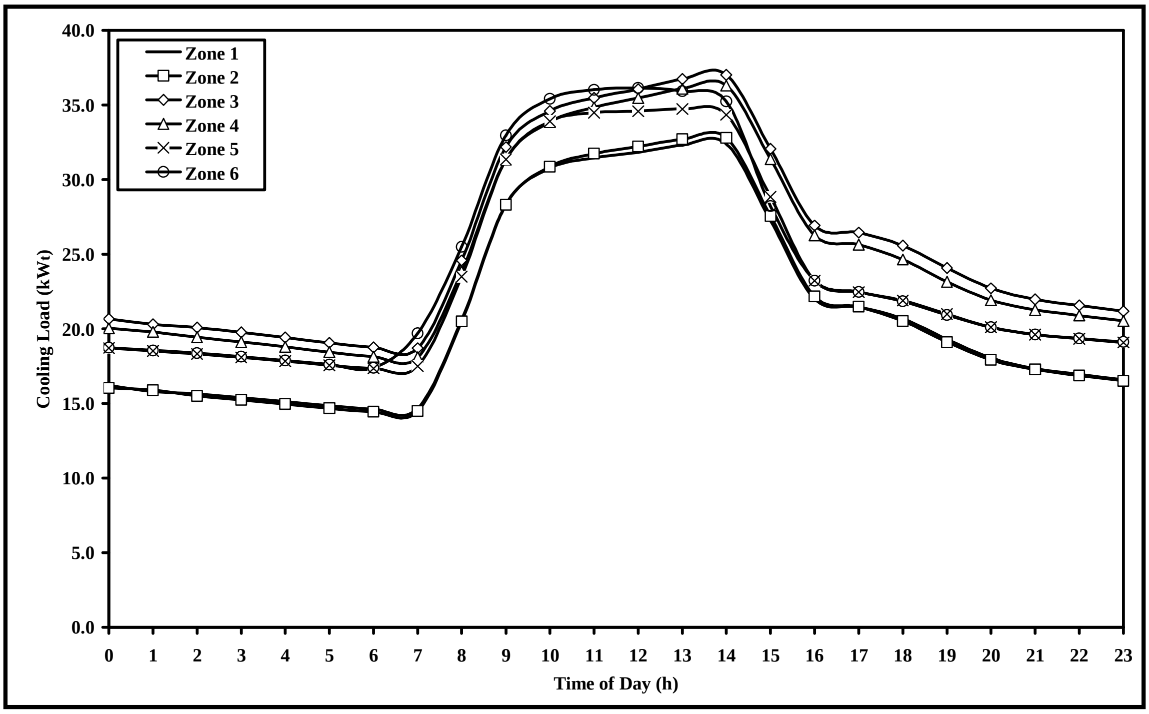

Figure 4.7 Average cooling load profiles for each zone in the clinic.

The maximum cooling load for zone number 3 is 36.3 kW_t and that for zone number 4 is 37.0 kW_t; both occur at about 14:00 hours, and the maximum cooling load in zones 3 and 4 are highest compared to other zones. The two zones are both located in the first floor of Block A of the building, their lower boundaries facing the upper boundaries of zones 1 and 2. The upper boundaries (that is, the roof) are facing the ambient conditions and they are directly subjected to the effect of the sun's rays. When the sun's rays strike the roof, some of the solar heat is absorbed by the roof and in conjunction with the high outdoor air temperature, this causes the surface temperature of the roof to rise even more than the ambient air temperature, therefore increasing the heat conduction transfer from the outer surface of the roof to the zones.

All zones in the building have nearly similar shape profiles because of the large ceiling area, however, Zones 5 and 6 have slightly different profiles. The zones are located at the east side of the clinic, so the walls facing the east are subjected to the solar incidence in the early morning when the sun starts to rise, and for that reason, the peak loads of the two zones are slightly earlier compared to other zones, and in the early hours at about 9:00 hours, the load in zone 5 approaches 90% of the maximum cooling load and that of zone 6 approaches 98%; this means the load is nearly flat from 9:00 hours to about 14:00 hours.

One method of reducing the solar heat gain on the external surfaces of the building is by using light coloured surfaces, since light colours reflect most of the solar radiation. Dark coloured external surfaces can absorb about 70% to 90% of incident solar radiation and therefore increase the surface temperature, hence increasing the cooling load of the building (Al-Juwayhel, 2003).

4.4.2.2 Whole building profile

The hourly average cooling load of the clinic is shown in Figure 4.8. The load was determined by adding up the individual building and ventilation loads of each zone. Figure 4.8 shows two bar plots of the cooling load versus time of day. The first plot is the hourly load of the building when there is no control over the ventilation. This, in fact, represents a common practice in most buildings in Kuwait where, in the summer season, the AC runs twenty four hours per day at a fixed temperature of about 24° C and without any control on the ventilation whether the buildings are occupied or not. Furthermore, sizing the AC systems incorporating cool thermal storage based on this load profile does not make this technology attractive especially when ice thermal storage technology is used, simply because the cooling load during the unoccupied period is still high. Therefore, there will not be enough cooling to charge the storage tank unless large chillers are used with a long charging time.

One method that can be applied to overcome the problem outlined above and to reduce the cooling load during the unoccupied period is to provide a control over the ventilation. In Kuwait, the ventilation load makes a significant contribution to the building cooling load because of the high dry bulb temperature and humidity even at night time, and for that reason, the AC systems cannot be switched off or even changed according to the zones, or the building temperature set to a value higher than 24° C unless the building is pre-cooled several hours before it is occupied. In an attempt to reduce the load during the night time without changing the temperature in the zone, the building was simulated again with a defined ventilation control that started at 5:00 hours and ended at 15:00 hours.

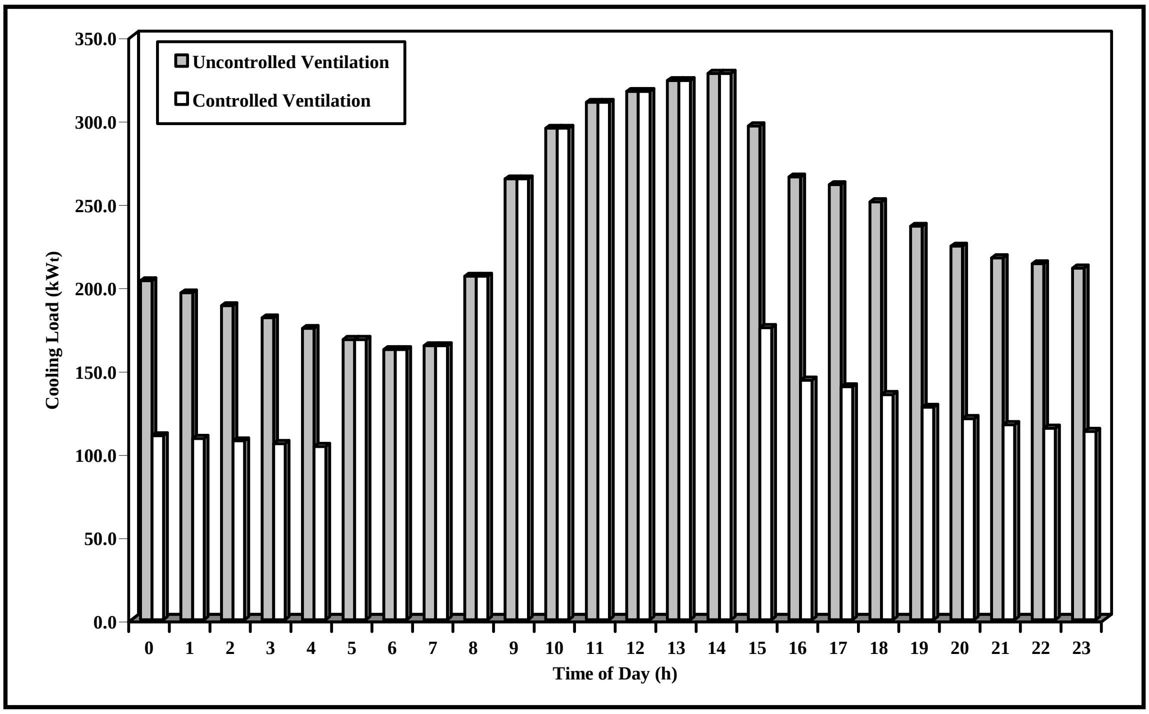

Figure 4.8 Cooling load profile of the clinic.

Building ventilation can be controlled by closing or opening the dampers in the AHUs. Since the clinic is not occupied at 15:00 hours, the fresh air dampers can be closed automatically or manually at this time, and they can be opened again at about 5:00 hours before the building is occupied at 7:00 hours to remove any contamination that may be present in the building and to provide a good and comfortable environment for the occupiers. The result of fresh air control is shown in Figure 4.8 above. This control helped to reduce the load not only at the night time but also at the peak load times between 15:00 hours and 17:00 hours. At 15:00 hours, the load was decreased from 296.3 kW_t to about 175.3 kW_t and this reduced the total integrated cooling load during the 24-hour period from 5.7 MW_th to 4.3 MW_th. The benefit of reducing the cooling load during the night time is to reduce the charging time of the cool thermal storage and reduce the size of the chillers.

ESP-r has already been used for calculating the hourly cooling load for different types of buildings in Kuwait (Al-sayed et al. 2001). Although these buildings have different occupancy, lighting, and appliances profiles, and different constructions and orientations compared to the clinic building, it is essential to look at their profiles to examine their variations, particularly at the peak cooling time, since this will help to select proper and general charging and discharging times when AC is designed with the use of a thermal storage system not only for the clinic building, but also for different types of buildings in Kuwait. Chapter 6 will discuss the selection of charging and discharging hours in more detail.

Figure 4.9 shows a normalised building cooling load for different types of buildings in Kuwait. The profiles were obtained using ESP-r building simulation, and their detailed analysis of the peak cooling loads and energy requirements are available in the study by (Al-sayed et al. 2001).

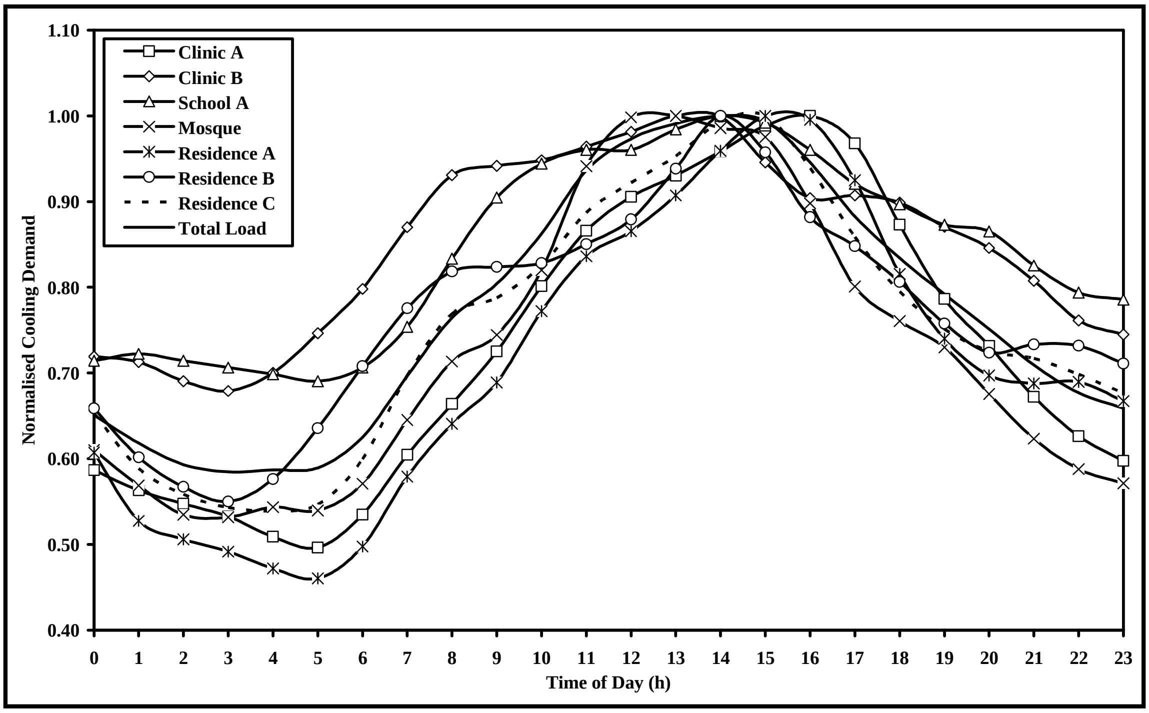

Figure 4.9 Normalised cooling load profiles for different types of buildings in Kuwait

For the residential buildings A, B and C, the occupancy profiles varied from 20% to 100%, lighting profiles varied from 50% to 100% and that for appliances from 0% to 100% depending on the type of rooms in the houses. It can be seen that for the residential buildings, the peak loads are slightly different because the orientation, windows, the wall constructions, and the number of people, lights, and appliances were different, and accordingly the maximum cooling load occurs between 14:00 hours and 16:00 hours.

For non-residential buildings such as clinic A, clinic B, school and mosque the peak in cooling load occurs between 12:00 hours and 16:00 hours as shown in Figure 4.9. It can be observed that the time at which peak cooling load occurs for most types of building in Kuwait ranges between 12:00 hours and 17:00 hours. Furthermore, the shape of the load profile of the clinic building under study looks similar to those developed profiles by (Al-sayed et al. 2001). The maximum cooling load of the clinic occurs at 14:00 hours which is between 12:00 hours and 17:00 hours as in the case for other buildings types.

In addition, because of the ventilation control that is applied to the clinic, the load at night is further reduced when compared to the loads of Figure 4.9 where there is no ventilation control.

The load profiles shown in Figure 4.9 were obtained without any ventilation control; applying a proper control on the ventilation would reduce the cooling load during the peak load period and reduce the cooling energy of the building even for buildings where the lights were on for 24 hours a day such as clinics A and B. With the cooling loads shown in Figure 4.9, thermal cool storage can be applied for most types of buildings in Kuwait. However, a proper ventilation control must be applied. In the case of hospital buildings, thermal cool storage may not be very attractive because of the high amount of fresh air required by these types of buildings for 24 hours a day. Nevertheless, this problem can be overcome by pre-cooling the fresh air that enters the AHU by using cooling recovery unit systems (Al-Ragom, 2003).

4.4.3 Cooling load breakdown

The proportion of the maximum cooling load of the building at peak time hour (for example, at 14:00 hours) is shown in Figure 4.10. The peak load is divided into five components: people, lighting, appliances, fresh air, and building structure. People, lighting, and appliances are usually grouped in a single category, namely, the internal cooling load. The cooling load in the building structure component includes the load due to window solar heat, glass conduction, and wall and roof conduction. The fresh air load depends on the amount of fresh air that is brought into the building through the mechanical fans present in the air handling units.

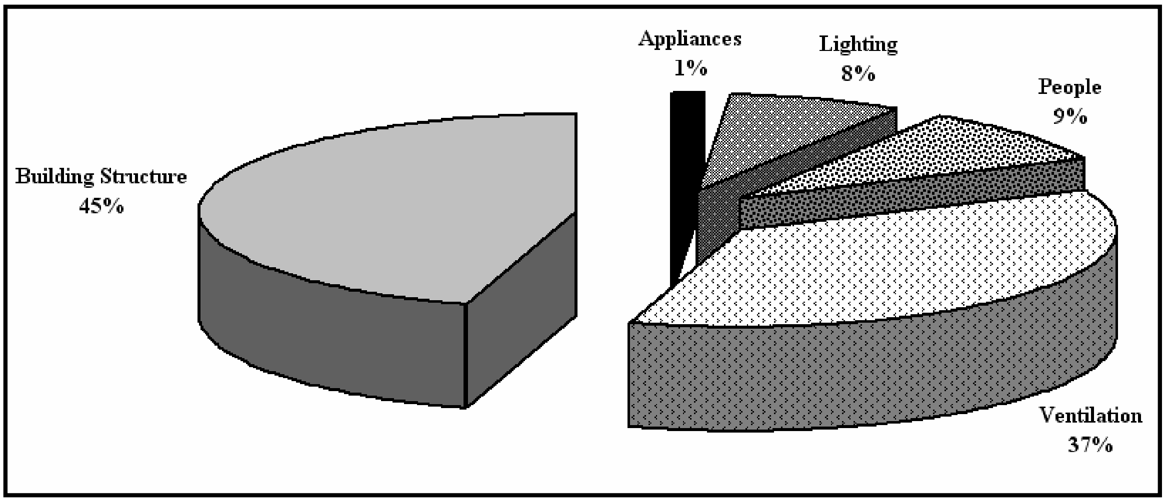

Figure 4.10 Breakdown of cooling load at peak time.

Figure 4.10 shows the proportion of each load component described above. The building structure has the largest heat gain and contributes about 46.0% of the total building load. Furthermore, it can be seen from the figure that the ventilation has also quite a large percentage of the load proportion, representing about 35.3%. The load from the building structure and fresh air are both a strong function of the external weather data; because of the extremely hot weather in Kuwait, their combined contribution is about 81.3% of the building peak load. The internal loads, people, lighting, and appliances have a total proportion of 18.6%.

4.4.4 Monthly cooling load

It is also important to examine the effect of ventilation control on the load profiles for each month in the year. The load profiles obtained from ESP-r for the months of March to November are plotted together and are shown in Figure 4.11 for uncontrolled ventilation and Figure 4.12 for controlled ventilation.

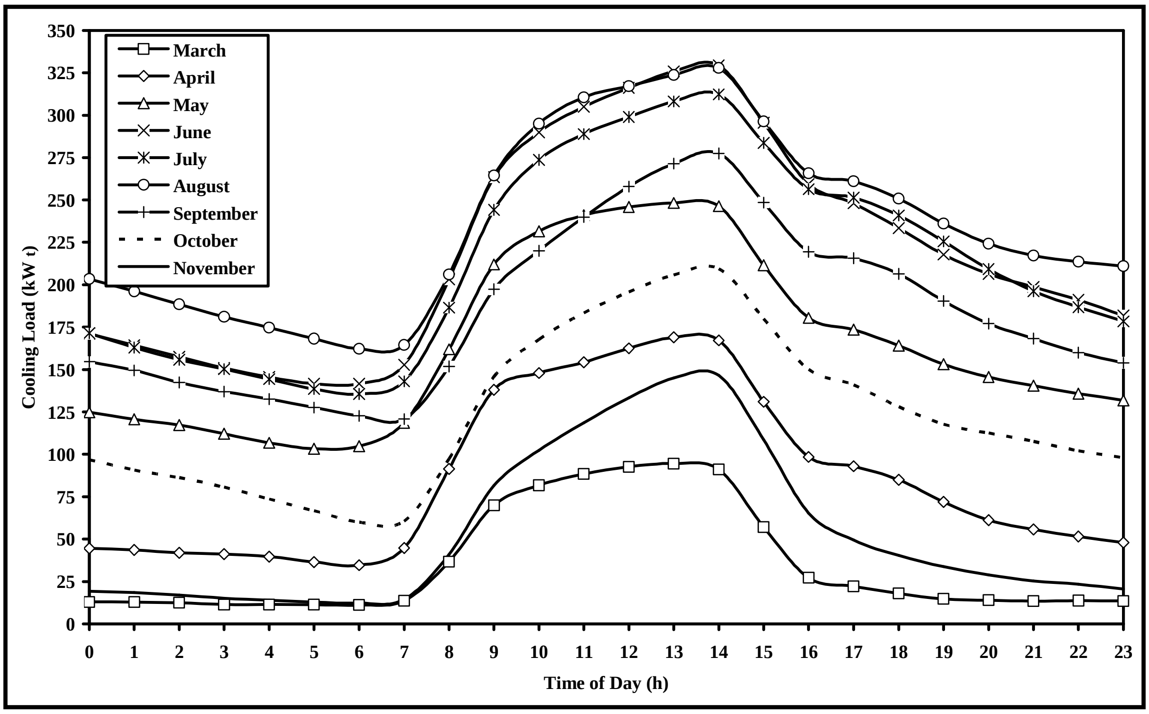

Figure 4.11 Average hourly cooling loads for March to November without controlled ventilation.

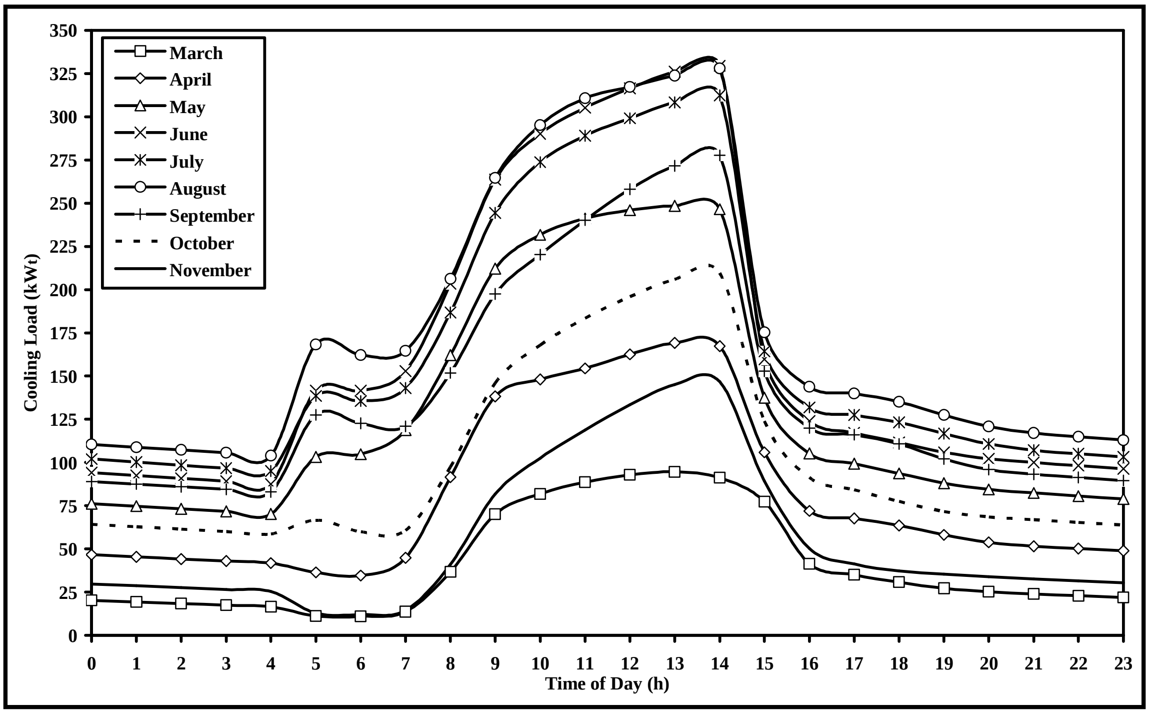

Figure 4.12 Average hourly cooling loads for March to November with controlled ventilation.

As discussed above, the fresh air flow into the clinic is controlled so that it starts at 5:00 hours, two hours before the building is occupied, and ends at 15:00 hours when the building is not occupied. Figures 4.11 and 4.12 are plotted to show the effect of this control on the building cooling load for the hourly average of March to November. By comparing these two figures, it can be seen that the ventilation control has the great advantage of reducing the building cooling load during unoccupied periods especially for the peak summer months from May to September, and also it has the good effect of lowering the load for other months such as October.

Figures 4.11 and 4.12 show how April and November are unlike the peak summer months. From the two figures, it can be observed that leaving the ventilation without any control can reduce the cooling load during the unoccupied period because in the early in the morning and late at night, the dry bulb temperatures are low, and it is better to supply the building with fresh air from outside continuously for 24 hours a day. Similarly, keeping the ventilation uncontrolled can help to reduce the cooling energy in March not only in the unoccupied period but also during the occupancy period. In March, the cooling load during the occupancy period can be further minimised by allowing more fresh air to enter the building and this can be simply done by further opening the fresh air dampers.

The hourly average cooling loads of January, February, and December are plotted in Figure 4.13 for uncontrolled ventilation and in Figure 4.14 for controlled ventilation. Figure 4.13 shows that the ventilation in the early morning (for example, from 12:00 hours to 16:00 hours) and at night (for example, from 18:00 hours to 23:00 hours) reduces the load in the building in those three months. This causes the temperature in the building to drop below the heating setting temperature of 21.0° C, which means the building needs additional heat to maintain the temperature at or above 21.0° C. This additional heat may not be really required, because if the building is not heated at this time the temperature will drop only slightly below 21.0°C and this will not cause a problem since the building is not occupied at this time of the day. When the ventilation is controlled, this prevents the heating requirement expect at the pre-cooling hours (for example, 5:00 hours to 7:00 hours) as shown in Figure 4.14.

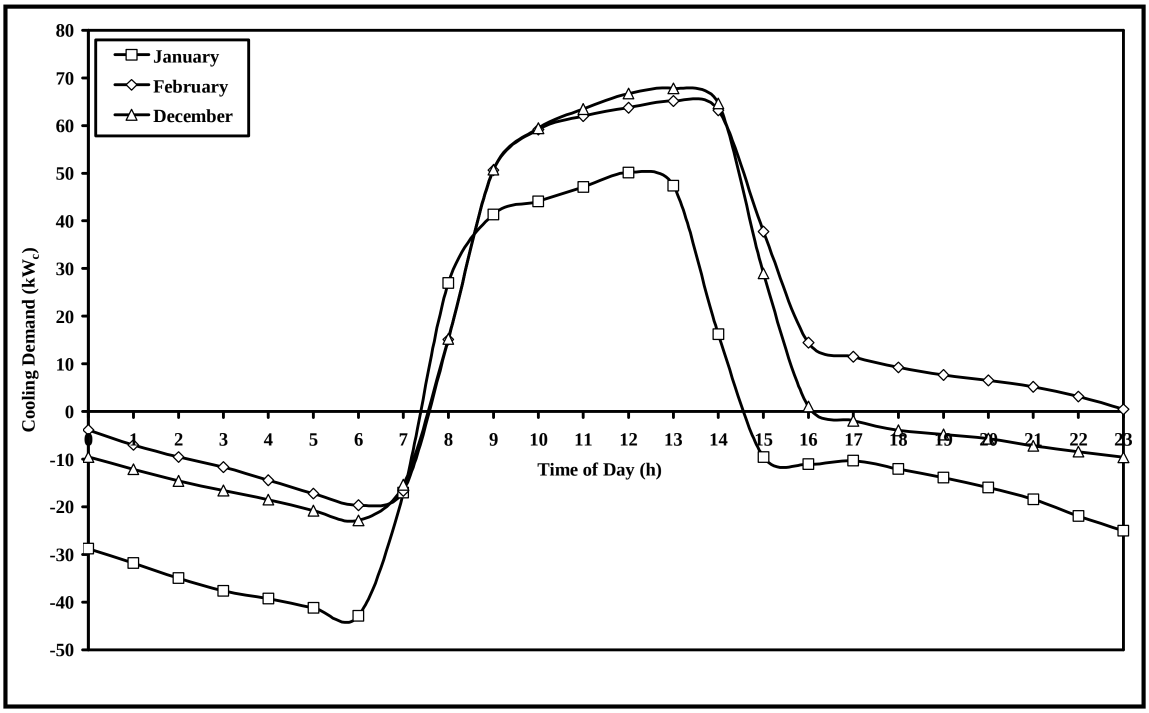

Figure 4.13 Average hourly cooling loads for January, February, and December without controlled ventilation.

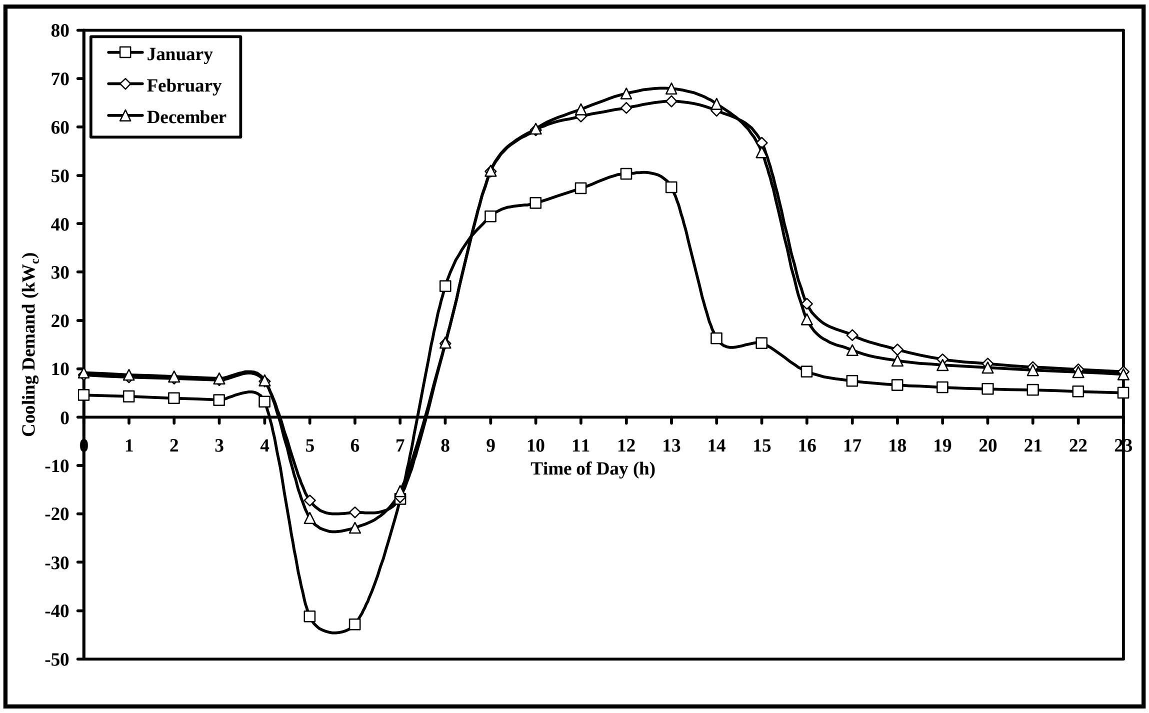

Figure 4.14 Average hourly cooling loads for January, February, and December with controlled ventilation.

During the occupancy period, January has the lowest cooling load. February and December have a slightly higher load compared to January. As in the case of March, the load can be reduced at the peak cooling time of these months by allowing more fresh air into the building. This will remove the additional heat gain in the building envelope without operating the chillers and pumps. However, it requires operating the AHUs with proper damper controls. The best way of controlling the ventilation at these months, specifically at the occupancy period time when the cooling load is high, is by fully opening the fresh air dampers and allowing more fresh air to enter the building as the temperature drops below 21.0° C in the building, the dampers can gradually be closed.

4.5 Model validation

The calibration process is one of the most difficult tasks in building energy simulation. Achieving high quality results can be difficult as well as time consuming. Unfortunately, the clinic was not equipped with any data loggers that could measure the hourly data for the individual components in the AC system. Thus, the lack of measurements and extra data availability often made it very difficult to obtain accurate hourly building cooling load data to be calibrated with the results of the energy simulation of the clinic.

A single data logger was installed in the clinic; its only function was to record the temperature measurements of the chilled water across the chillers, ice thermal storage, and AHUs. The purpose of taking the temperature measurements was to obtain the cooling production from the chiller during charging and discharging periods, and to obtain the amount of cooling production from the storage tanks during the peak load period. The cooling production for the chiller for a few days was determined for August 2003 and when the system was running conventionally (Maheshwari, 2005).

The power inputs of the motors for pumps, and AHUs using a power meter were measured and the heat gains by the air distribution ducts and water distribution pipes had been already calculated by the HVAC designer of the building and were made available (BLK, 2000). Since the motors are located in an air conditioned space, all electrical input power is converted to heat and this heat places an extra load on the chiller. The actual hourly building cooling load of the clinic was determined by subtracting the heat gain by all motors, pipes and ducts from the cooling production of the chiller.

From the measurements of the input power, the total actual system heat gain was determined to be 66.7 $kW_{t;}$ it is constant throughout the year because constant speed motors were used. The actual average peak cooling production in August at 14:00 hours was estimated to be about 368.4 kW_t. This value was obtained based on the available data taken at dates 11th, 12th, and 22nd of August 2003 (Maheshwari, 2005), and when the system at that period of time was running conventionally. Therefore, the actual cooling load of the clinic at 14:00 hours was calculated to be 301.7 kW_t. ESP-r predicted the load at 14:00 hours to be about 327.9 $kW_{t,}$ which is 8.0% higher. However, the predicted daily integrated cooling load from ESP-r is lower than the actual integrated load by 8.9% and is equal to 5.7 MW_th.

The above simple comparison was done for only three days in August. For a more detailed model calibration, each specific element of the model would have to be calibrated against known data (Thomas, 1999). In a very hot climate such as Kuwait, external conditions (specifically weather data) drive the energy use of the building, and thus it follows that calibration of only the external load can produce an accurate model. One of the difficulties that may be encountered in the simulation of building energy is the estimation of the ventilation load. The ventilation load mainly depends on the external weather and internal building conditions. ESP-r (Clarke, 2001) treats the ventilation as infiltration and uses two correlations for average permeability and tight construction buildings, which are a function of wind speed and temperature difference. Neither correlation plays any part within the building simulation program, nor has any physical reality (Grenvill, 1994). The internal conditions can be better estimated by determining the cooling coil performance in the simulation program, and this requires simulating the building as well as the cooling system.

In mild climates such as Hong Kong (Li, 2003) the internal load dominates, which means the heat gain contribution of occupants, lighting, and equipment is greater than the contribution of the solar isolation and window and wall conduction. Therefore, more effort must be spent to account accurately for the heat gain on the internal loads these efforts are discussed by (Thomas,1999; Pedrini, 2001).

4.6 Conclusions

The building energy simulation program, ESP-r, was used to determine the cooling load of the clinic building for the design day and whole year using the typical metrological weather data for Kuwait. The simulation was carried out at KISR and conducted by defining the geometry and the construction of the internal and external walls, roof, ceiling, and ground floor, the boundary conditions for each surface, and the operation schedules of the internal load such as occupancy, lighting, and appliances for each zone of the clinic.

The results were analysed for individual zones. The profiles for the building cooling load were determined with and without ventilation control. For zones number 1, 2, 3 and 4, the maximum cooling demand occurred at 14:00 hours, and for zones number 5 and 6 occurred slightly earlier at 13:00 hours. Furthermore, ventilation control showed a significant reduction in the cooling load especially for peak summer months during the unoccupied period. This makes the use of cool thermal storage more attractive because of the lower load during the time at which the storage is charged.

The hourly average cooling load with controlled ventilation for August was selected for sizing the components of the AC systems for both conventional and cool thermal storage systems. The maximum cooling load of the building was estimated to be 328 kW_t and the total integrated building load was 5.7 MW_th.

Moreover, the cooling load profiles determined by using ESP-r for different types of buildings including residence houses, clinics, mosques, and schools in Kuwait were examined. The analysis showed that the maximum cooling load always falls between 12:00 hours and 17:00 hours for most types of buildings in Kuwait. The time at which the maximum cooling load of the clinic building was occurred at 14:00 hours which was found to be in the range for most types of building in Kuwait.

The simulated average peak and total integrated cooling load of August was compared with the actual measurement load. The results showed that the peak cooling load obtained from ESP-r is higher than the actual value by 8.0%, and the total integrated load is lower by 8.9%. The higher estimation of the maximum cooling load by ESP-r might be resulted from not accounting shutters of the windows in the clinic building. Because of unavailable actual data for the whole year, the comparisons for other months have not been performed and therefore have not been given and discussed in this chapter.

Chapter 5

Calculation of System Load

5.1 Introduction

In Chapter 4 it was shown how ESP-r was used to determine the profile of the cooling load of the clinic due to the internal loads such as sensible loads from lights, appliances, and occupants, and latent loads mostly from occupants, envelope loads due to heat transmission through walls, windows, roofs, floors and slabs, together with solar gains through fenestration and ventilation loads. However, there are other loads resulting from the operation of the AC system that must be added to the building cooling load; these are known as auxiliary system loads. The auxiliary system loads are generated as a consequence of heat gains by

a. AHU fan motors.

b. Chilled water pump motors.

c. Ducts

d. Pipes

In order to determine the heat gains generated by the auxiliary systems, each component of the AC system must be sized adequately. This chapter illustrates the sizing procedure of each component, for example, chillers, storage, pumps and so on, in conventional, ice storage, and chilled water storage AC systems, and then calculates the loads generated from the heat gain by the auxiliary systems. The calculated auxiliary loads will then be added to the building cooling load to obtain the system load profiles for all AC systems.

The procedure of designing, sizing and selecting of the various components for conventional, ice and chilled water storage AC systems developed in this chapter can be used for any air-cooled AC systems for medium size buildings in Kuwait. However, for water-cooled systems additional information required for sizing and selecting of the cooling tower and condenser water pumps.

Moreover, the information given in this chapter regarding the sizing of the AC components will further help to study the energy performance of each AC system as will be shown in the next chapter; also the information in this chapter will help to estimate the capital cost of the AC systems in order to conduct the life cycle cost analysis, as will be further illustrated in Chapter 7.

5.2 System load estimate

The profile of the building cooling load for the clinic was determined using the building energy simulation program, ESP-r, as discussed in Chapter 4, and is shown in Figure 4.8 with controlled ventilation. The initial estimation of the system load for the building was obtained by adding the heat gain generated from the auxiliary systems to the building cooling load.

The heat gains of the auxiliary systems were obtained based on the assumptions that were made for the efficiency of fans, pumps and motors and the heat gain by the piping systems. The initial estimation of the heat gain by the AHU fan motors was calculated based on a motor efficiency of 80%, total fan efficiency of 60%, the calculated volumetric flow rate of air as given in appendix A, and a total pressure drop of 0.9 kPa.

The total pressure drop of 896 Pa was recommended by different AC consultant offices (El-Amer, 2005; Bulos, 2005) for the initial design stage for the ducting system; it was on this basis that the ducting system of the clinic building was classified as a medium pressure system having a maximum pressure drop of up to 1.0 kPa (Jones, 1994). The assumed total pressure drop was obtained based on the pressure drop in the duct being 0.4 kPa and the pressure drop in the AHU being 0.5 kPa.

Similarly, the initial estimation of the heat gain by the secondary chilled water pumps was obtained by assuming the pumps had 60% efficiency and the motors had 80%. The pressure drops and the volumetric flow rates were computed as shown in Section 5.4. However, for the primary pumps the heat gain could not be estimated at the initial design stage because the flow rates were still unknown; therefore, their values were guessed as a percentage of the peak system load of the building. For most of the AC systems of buildings in Kuwait the average heat gain by the auxiliary systems at the peak building load is approximately 17.5% higher than the building cooling load depending on the efficiencies of the AHUs and pumps motors and the insulations of ducts and pipes of the AC system (Maheshwari et al. 2003). Therefore, the system load can be approximated by multiplying 1.175 by the building cooling load obtained from the ESP-r building simulation. A reasonable estimate of the heat gain by the primary pumps was taken as 2% for the conventional and chilled water storage AC systems and 6% for the ice storage systems to account for the extra pressure drop in the ice tanks. In a similar manner, the heat gain by the piping systems was also estimated as a percentage of the peak system load of the building. The heat gain was approximately 2% (Maheshwari et al. 2003) and this value was used for the conventional and chilled water storage AC systems, and the slightly higher value of 3% for the ice storage AC system because of the lower chilled water temperature circulation in the systems. The profile of the system load is shown in Figure 5.1 for the conventional, ice, and chilled water storage systems.

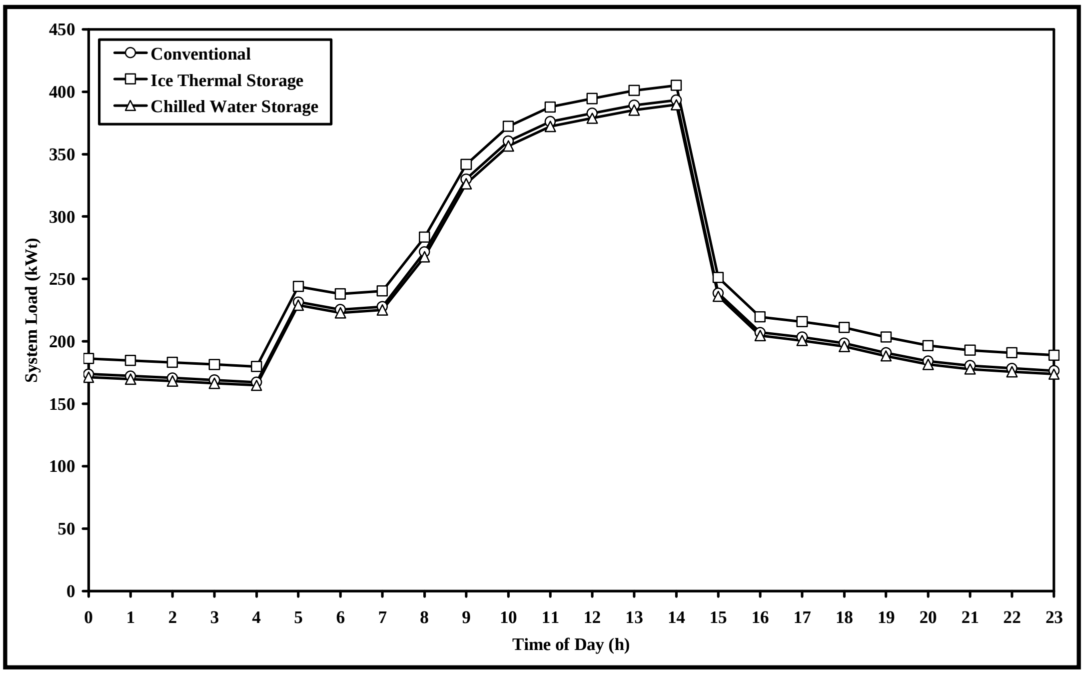

Figure 5.1 Assumed system load of conventional, ice storage and chilled water storage AC systems for August.

The ice and chilled water system load profiles, as shown in Figure 5.1, were obtained for the full storage operation strategy and can be applicable for sizing the chiller and storage capacities for load levelling and 50% demand limiting partial and full storage systems since the building cooling load profiles for all operation strategies are the same and the only difference is the heat gain by the primary pumps, which is small.

Referring to Figure 5.1, the profiles for the conventional ice and chilled water storage systems differ slightly. This is due to the difference in the assumptions that were made for the calculation of heat gain by the primary pumps and pipes and in the estimations that were made for the secondary pumps. Furthermore, the figure shows that the ice storage system has a slightly higher cooling load compared to conventional and chilled water storage systems, because of the slight increase in the percentage assumptions of the heat gain by the primary pumps and additional heat gain by the piping systems.

For all profiles, the maximum cooling load occurs at 14:00 hours, and the minimum occurs at about 16:00 hours. The integrated system load for the conventional is 5.6 MW_th, ice storage is 5.9 MW_th and chilled water storage is 5.6 MW_th. The diversity of the conventional and chilled water storage is 0.61 and for ice storage is 0.63. The diversity is defined as the ratio of average to peak system load.

The system loads profiles in Figure 5.1, which are based on the assumed heat gain by the auxiliary systems, will be compared later with a properly calculated heat gain by the auxiliary systems to ensure that the peak cooling load and the total integrated cooling load are higher than those in the figure. Otherwise, the heat gain by the auxiliaries must be recalculated with different assumptions. This will be done after a proper determination of the sizes of each component in the AC systems.

5.3 Operation strategies

The operation strategy of ice and chilled water thermal storage systems has a significant effect on the capacity of the chiller and the storage and, hence, on their capital costs. Furthermore, the electrical energy consumption of the chillers is strongly dependent on the selection of the operating strategy of the system (Simmonds, 1994). Different operating strategies can be examined and compared to a conventional AC system. These operating strategies have already been discussed in section 3.4 of Chapter 3.

In this report, partial and full storage strategies are examined for both ice and chilled water storage systems. For the partial storage strategy, load levelling (with chiller priority strategy) and 50% demand limiting strategies are examined. The advantage of using chiller priority over storage priority strategy is that with chiller priority both the chiller and storage are smaller; this could result in lower capital costs. In addition, with chiller priority the control system and sequences are more complicated. However, the only advantage of storage priority is consume slightly lower energy (Simmonds; 1994). In the chiller priority strategy, the chiller meets the cooling load as much as possible and when the cooling load exceeds the chiller capacity, the additional cooling is supplied from the storage tank. In the full storage strategy, the chiller switches off completely during peak cooling load (for example, the discharging cycle) and the stored ice or water meets the cooling load.

5.4 Charging and discharging times

Selecting the charging and discharging period is one of many factors that can be controlled by the system designer or operator (Beggs, 1992). For many countries such as the UK and the US, the charging and discharging time is strongly related to the electricity tariff structure; usually in these countries, the charging time is selected so that it coincides with when the cost of electricity is low (Beggs, 1992). However, in Kuwait, the case is different because the cost of electricity does not vary during a 24 hour period. Moreover, the capacities of both the chiller and ice or chilled water store are very much dependent on the duration of the charging time. The longer the charging time, the greater the capacity of the store can become, so the output of the chiller can be reduced. The time at which the ice and chilled water is charged and discharged must be carefully selected if the system is to be designed for partial demand limiting and full storage operation strategies.

The main aim of using ice and chilled water storage AC systems is to reduce the electrical demand on the building and therefore on the power station during the peak electrical demand, hence the selection of the charging and discharging time must be considered based on the profile of the system load (for example, building plus auxiliary) as well as the profile of the electricity power generation of Kuwait. To make the correct decision for selecting the charging and discharging time, the profiles of the system load and the electricity generation are plotted together as shown in Figure 5.2.

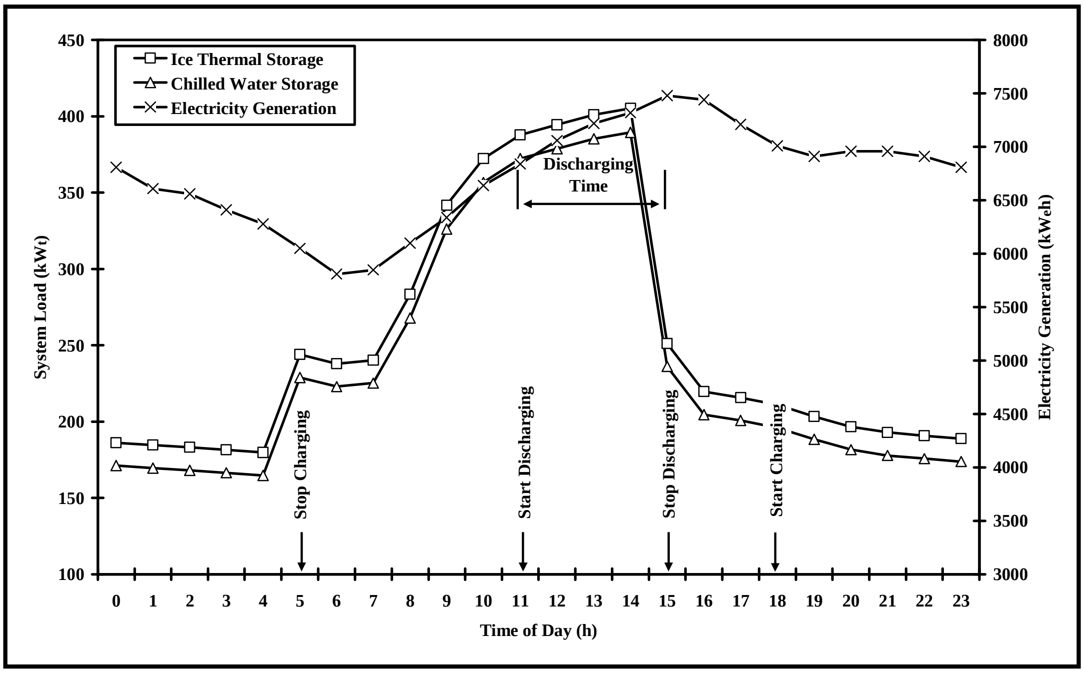

Figure 5.2 Selecting charging and discharging hours for ice and chilled water storage AC systems for August.

It can be seen from the figure that at about 16:00 hours the system load was low because the ventilation load was reduced at this time of the day by closing the fresh air dampers as discussed in Section 4.6.2 of Chapter 4; however, the charging of the storage tank must be avoided at this time since the demand for electricity in the country is still high. Storage discharging can be taking place from 11:00 hours to 15:00 hours because during this time the system load and electricity demand are both high, thereby reducing the electrical demand on both the building and power stations.

The charging time can be selected as being any where between 18:00 hours and 5:00 hours; the choice of a longer charging time means the chiller capacity will be reduced. Generally, the choice of the charging and discharging time depends upon a given cooling load profile. If a building has a different load profile than the one shown in Figure 5.2, then the charging and discharging times will change accordingly.

5.5 Component sizing

Appropriate sizing of AC components is critical to the design of energy-efficient AC systems. Correctly-sized systems cost less to install, operate, and maintain. Undersized systems will not meet a space's temperature, humidity, or indoor air quality requirements. Oversized systems cost more to install, need more space, and operate less efficiently in part-load conditions.

This section discusses the sizing procedure of each component including chillers, ice and chilled water storages, and AHUs and pumps in the conventional, ice storage, and chilled water storage AC systems. Having obtained the correct sizes of the components for each AC system, the capital cost of the systems can be obtained; hence, the life cycle cost can be performed as will be shown later in Chapter 7.

5.5.1 Chillers

Using the system load profiles of the clinic developed in Section 5.2 and shown in Figure 5.1, and applying the selected charging and discharging time as discussed in Section 5.4, the size of the chiller capacity for each operation strategy (for example, load levelling and 50% demand limiting partial and full storage) was obtained for the ice and chilled water storage systems by applying the following equation (Dorgan, 1994)

$$C_{chil} = \frac{\text{Integrated system load}(kW_t h)}{H_{char} CR_{char} + H_{dirt} CR_{dirt}} \tag{5.1}$$

At this stage, the chiller capacity C_{chil} is unknown; however, the capacities relative to some standard conditions can be identified for each hour. The chiller capacity C_{chil} may actually vary depending upon the condenser and evaporator conditions, but these can be reduced to two conditions according to whether the storage tank is charged simultaneously with the cooling of the building at night time, or only with direct cooling at day or night times.

In equation (5.1), the chiller capacity ratio CR is defined as the ratio of the cooling production of the chiller at any hour to the chiller nominal capacity. It can be expressed for each cooling mode as a percentage of its nominal capacity. Nominal capacity must be selected as the capacity, for example, at the rating conditions of the Air Conditioning Refrigeration Institute (ARI) US or any other accepted standards given in (MEW, 1999b), provided that necessary correction and provisions are made to suit Kuwait.

The chilled water leaving the chiller in an ice storage system during the charging mode is usually in the range of −3.0 to −6.0° C, and the capacity ratio CR_{char} at these temperatures is in the range of 0.6–0.7 of the nominal chiller capacity (Dorgan, 1994). During direct cooling, when the outlet temperature from the chiller is about 5.56° C, the capacity ratio CR_{dirt} is 1.0. Therefore, the capacity ratio when charging CR_{char} was chosen as 0.65, and when direct cooling CR_{dirt} was chosen as 1.0 of the nominal chiller capacity. This means a 1.0 MW_t capacity chiller will provide 0.65 MW_t cooling during the charging mode and 1.0 MW_t cooling during the direct cooling mode.

For the chilled water storage system, the typical chilled water temperature leaving the chiller during the period of charging, discharging, and direct cooling is usually in the range of 4.0° C to 6.67° C. Therefore, for the sizing of the chiller capacity, the capacity ratios when charging CR_{char} and when direct cooling CR_{dirt} of the chiller were assumed equal to 1 and similar to a conventional AC system.

Moreover, equation (5.1) was also used to obtain the nominal chiller capacity C_{chil} for a conventional AC system. When applying equation (5.1) to determine the chiller capacities for all AC systems, the calculated nominal chiller size C_{chil} was either greater or smaller than the cooling load during direct cooling, therefore the chiller size was recalculated and adjusted by solving the equation with the required iterations in order to determine properly the correct nominal chiller size C_{chil}. The computed nominal chiller sizes in the conventional, ice, and chilled water storage AC systems are summarised in Table 5.1.

Air Conditioning System	Nominal Chiller Size (kW_t)
Conventional	393
Ice Storage (Partial Load Levelling)	336
Ice Storage (Partial 50% Demand Limiting)	406
Ice Storage (Full)	532
Water Storage (Partial Load Levelling)	242
Water Storage (Partial 50% Demand Limiting)	283
Water Storage (Full)	328

Table 5.1 Calculated nominal chiller sizes for the AC systems

ECAT2 Version 4.12 (Carrier, 2004) chiller selection software was used to select suitable air-cooled screw chillers using the nominal chiller capacities given in Table 5.1. ECAT2 provides performance data based upon a factory run test for each model of chiller at the full load design point and at the part load operating points as required.

The screw chillers for the different operating strategies were selected based on the following input data,

a. Fluid type.

b. Nominal chiller capacity in W_t

c. Fluid volumetric flow rate in m^3s^{-1}

d. Temperature of the fluid entering the chiller in °C

e. Temperature of the fluid leaving the chiller in °C

f. Fouling factor for the coolers in $m^2°KW^{-1}$

The fluid type for conventional and chilled water storage systems is water, and in the ice storage system is a 25% mass ethylene glycol mixture. Furthermore, the nominal chiller capacities are given in Table 5.1. The fouling factor for the cooler and the design dry bulb temperature of the air entering the condenser for the Kuwait coastal area were 0.4403 $cm^2°KW^{-1}$ (MEW, 1999b) and 47.4° C respectively (Shaban, 2000).

For the conventional AC system, the recommended temperatures of the chilled water entering and leaving the chiller were taken as 6.7° C and 12.8° C respectively; for the chillers in ice storage AC systems, the entering temperatures were 11.67° C and leaving were 5.56° C during normal mode; and entering temperatures were -5.0° C and leaving were -1.0° C during charging mode. Furthermore, in the chilled water storage AC systems, the water distribution systems are usually designed with a higher temperature differential to reduce the size of the volume of the storage tank; therefore, the temperatures of the chilled water entering and leaving the chiller were taken as 5.56° C and 15.56° C respectively (refer to Appendix B for the further chiller specifications).

5.5.2 Storage

The storage size or capacity in (kW_th) for each operation strategy was computed by Dorgan (1994)

$$\text{Storage capacity}(kW_t h) = C_{chil} H_{char} CR_{char} - TC_{char} \tag{5.2}$$

Equation (5.2) means that the storage capacity in (kW_th) is equal to the chiller capacity minus the total integrated cooling load of the building at charging hours H_{char} only. If the system load during the charging hours H_{char} is zero, then the chiller will charge only the storage tank, and the second term in equation (5.2) becomes

$$\text{Storage capacity}(kW_t h) = C_{chil} H_{char} CR_{char} \tag{5.3}$$

Equation (5.2) was used to calculate storage capacity in both ice and chilled water storage AC systems. For chilled water storage AC systems, the volume of the water tank is additional information that must be determined. Thus, the storage capacity obtained from equation (5.2) was used to determine the volume of the tank (Dorgan, 1994), for example,

$$V = \frac{3600 \times \text{Storage capacity}(kW_t h)}{\rho c_p (T_i - T_o)(\text{figure of merit})} \tag{5.4}$$

For most storage tanks, the temperature of the discharge ranges between 4.4°C and 7.0° C, and the return ranges from 10.0° C to 18.0° C (Dorgan, 1994; William, 2003). A storage tank with a good diffuser design can perform at a figure of merit of 90% or better (Dorgan, 1994). This means if the chiller stores 100 kW_th of cooling energy in a storage tank, only 90 kW_th of cooling can be withdrawn from the tank and 10 kW_th is lost due to the mixing effect within the storage tank and heat transfer to the chilled water across the storage boundaries from the surroundings. The figure of merit can be accurately estimated only from field data or from scale model testing of similar geometries. In the absence of such data, a factor of 0.85 to 0.90 can be used as proposed by Dorgan (1994). The volume of the storage tank for each operation strategy in chilled water storage AC systems was obtained using equation (5.4) based on a figure of merit of 0.9, an average discharge temperature of 6.67° C, and return water temperature of 16.67° C. Table 5.2 shows the calculated storage capacities of different operation strategies for ice and chilled water storage systems including the volume of the tanks in the chilled water storage systems.

Air Conditioning System	Storage Capacity (kW_th)	Storage Volume (m^3)
Conventional	0	0
Ice Storage (Partial Load Levelling)	272	0
Ice Storage (Partial 50% Demand Limiting)	714	0
Ice Storage (Full)	1527	0
Water Storage (Partial Load Levelling)	785	76
Water Storage (Partial 50% Demand Limiting)	1056	102
Water Storage (Full)	1514	146

Table 5.2 Storage capacities for the ice and chilled water storage systems

Using the calculated storage capacities of the ice storage systems, the specifications (see Appendix B) including pressure drop, dimensions, and the total storage capacity (latent plus sensible) and the required number of ice tanks per system were obtained from available catalogues from two different models from two different suppliers (DUNHAM-BUSH, 1995; CALMAC, 2001a).

For the chilled water storage systems, further details of the water tanks such as shape, dimensions, and diffuser designs were obtained. It was assumed that the water tanks have a cylindrical shape and based on this, the dimensions of the tanks were calculated (see Appendix C). Furthermore, the diffuser design associated with each operation strategy is discussed in detail in Appendix C and some important characteristics of these diffusers are given in Table 5.3.

Diffuser Description	Partial Storage		Full Storage
	Load Levelling	50% Demand Limiting	
Maximum Volumetric Flow Rate (m^3s^{-1})	5.41×10^{-3}	7.42×10^{-3}	9.30×10^{-3}
Assumed Reynolds Number Re_i	850	850	850
Inlet Water Kinematic Viscousity (m^2s^{-1})	1.50×10^{-6}	1.50×10^{-6}	1.45×10^{-6}
Volume Flow Rate Per Unit Diffuser Length (m^2s^{-1})	0.00127	0.00127	0.00123
Total Diffuser Length (m)	4.3	5.8	7.6
Length of Each Side of the Octagon Diffuser (m)	0.5	0.7	0.9
Assumed Froude Number	0.7	0.7	0.7
Acceleration of Gravity (ms^{-2})	9.81	9.81	9.81
Density of Inlet Water (kgm^{-3})	999.8	999.8	999.8
Density of ambient Water (kgm^{-3})	998.7	998.7	998.7
let Opening Height (cm)	6.7	6.7	6.6

Table 5.3 Diffuser characteristics for each design operating strategy in a chilled water storage system

The diffusers are the most important elements in the design of the chilled water tanks in chilled water AC systems. It is essential to know the description of the diffuser pipes in order to determine correctly the pressure drop and therefore properly calculate the total heat gain required by the primary and secondary pumps as described in Appendix D. Appendix C illustrates in more detail the sizing and the design of the diffusers' piping network for the chilled water tanks that were obtained for the load levelling partial, 50% demand limiting partial, and full storage chilled water systems.

In Table 5.3, the volume flow rate per unit diffuser length, q, was first computed based on an assumed inlet Reynolds number Re_i of 850 (Dorgan, 1994) and then the total diffuser length L was calculated. The length of each section of pipe of the octagon diffuser was obtained by dividing the total diffuser length L by 8. Based on the Froude number Fr_i of 0.7, which is less than 1 as recommended by (Yoo, 1986) and (Willden, 1989), the inlet height h_i was obtained. Furthermore, the density of the inlet water ρ_i was calculated at a temperature of 5.56˚ C and that of the ambient water ρ_w was calculated at 16.67˚ C.

The characteristic of the pipe section in the octagon diffuser was further obtained based on the maximum allowable velocity in the diffuser pipe. Table 5.4 provides a description of one side section of the octagon diffuser pipe in chilled water thermal storage systems.

Pipe Description	Partial Storage		Full Storage
	Load Levelling	50% Demand Limiting	
Diffuser Pipe Nominal Diameter (mm)	80	100	100
Diffuser Pipe Internal Diameter (cm)	7.79	10.23	10.23
Diffuser Pipe External Diameter (cm)	8.89	11.43	11.43
Radial Length (120°) (cm)	8.16	10.71	10.71
Total Area of the Branch (cm^2)	47.7	82.1	82.1
Total Area of the Slots (cm^2)	21.7	31.6	37.3
Assumed Number of Slots	8	9	10
Flow Rate per Slot (cm^3s^{-1})	85	103	116
Slot Area (cm^2)	2.7	3.5	3.7
Slot Width (mm)	3.3	3.3	3.5
Maximum Flow Velocity at the Inlet to the Octagon (ms^{-1})	0.284	0.226	0.283
Maximum Flow Velocity in the Diffuser Pipe (ms^{-1})	0.142	0.113	0.142
Maximum Velocity Outlet Opening (ms^{-1})	0.312	0.294	0.311

Table 5.4 Characteristics of a single section pipe in the octagon diffuser

The nominal size of the diffuser pipe was obtained based on the allowable maximum velocity in the diffusers of 0.274 ms^{-1} as recommended by (Fiorino, 1991). As shown in Table 5.4, the calculated maximum velocity in the diffusers for all operation strategies was lower than 0.274 ms^{-1}. Furthermore, Fiorino (1991) argued that lowering the velocity in the diffuser precludes a turbulent, jet-like flow near the diffuser openings.

Furthermore, the total slot area in the diffuser was calculated so that it was equal to or less than half of the cross sectional area of the inlet diffuser pipe to reduce the viscous pressure drop along the branch pipe compared to the pressure drop across a representative opening. This tends to reduce the variation in flow rate through successive openings along a diffuser branch because the flow rate through an opening depends directly on the pressure drop across it (Wildin, 1990). The total slot area was modified to satisfy the condition that the recommended maximum velocity at the slot opening be in the range of 0.3 ms^{-1} to 0.6 ms^{-1} (Dorgan, 1994). Moreover, based on an assumed number of slots, the slot cross sectional area for each slot was calculated; hence, the slot width was obtained as shown in Table 5.4 above.

5.5.3 Further discussion

To understand further the use of equations (5.1) and (5.2) for sizing the chillers and storages, Figures 5.3 - 5.5 were generated, which illustrate the profiles of the cooling production of the chillers operating in conventional, ice storage, and chilled water storage AC systems respectively. Furthermore, for ice and chilled water storage AC systems, Figures 5.4 and 5.5 show the cooling production of the chillers in load levelling, 50% demand limiting partial storage, and full storage operation strategies.

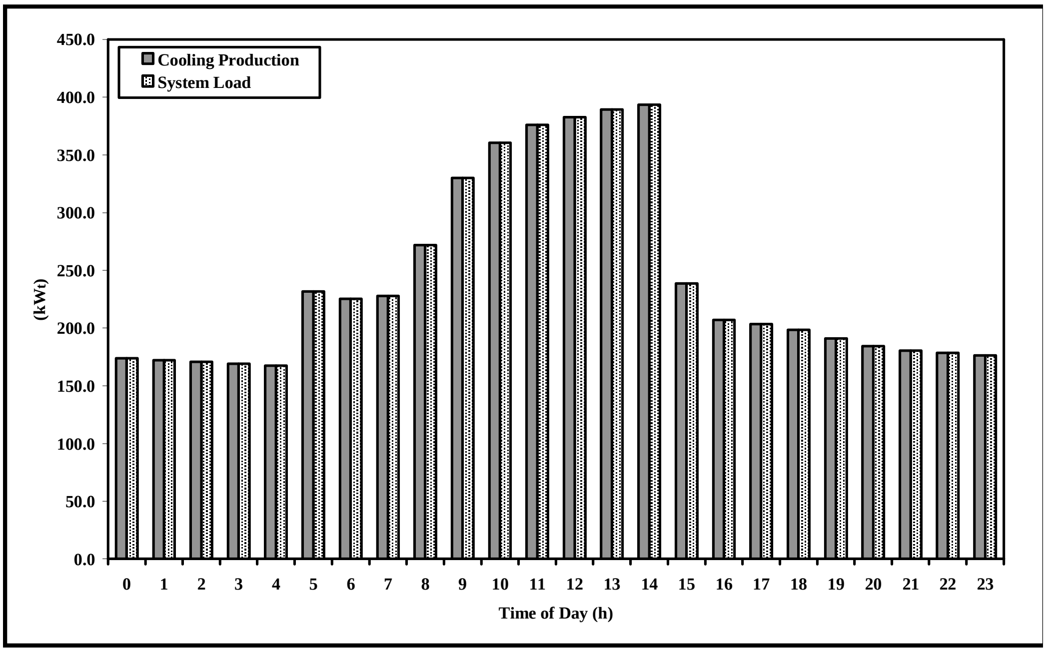

Figure 5.3 Hourly profile of cooling production of the chiller in conventional AC system.

Figure 5.3 shows a bar plot of the system load and cooling production of the chiller versus time of day operating in a conventional AC system. As can be seen from the figure for a conventional AC system operation, the chiller operates during 24 hours to provide sufficient cooling to meet the cooling load of the building, and for most of the time, the chiller operates at part load. Although equation (5.1) was used to determine the chiller capacity, the chiller can be obtained directly by referring to the maximum system load of 393.4 $kW_{t,}$ which occurs at 14:00 hours.

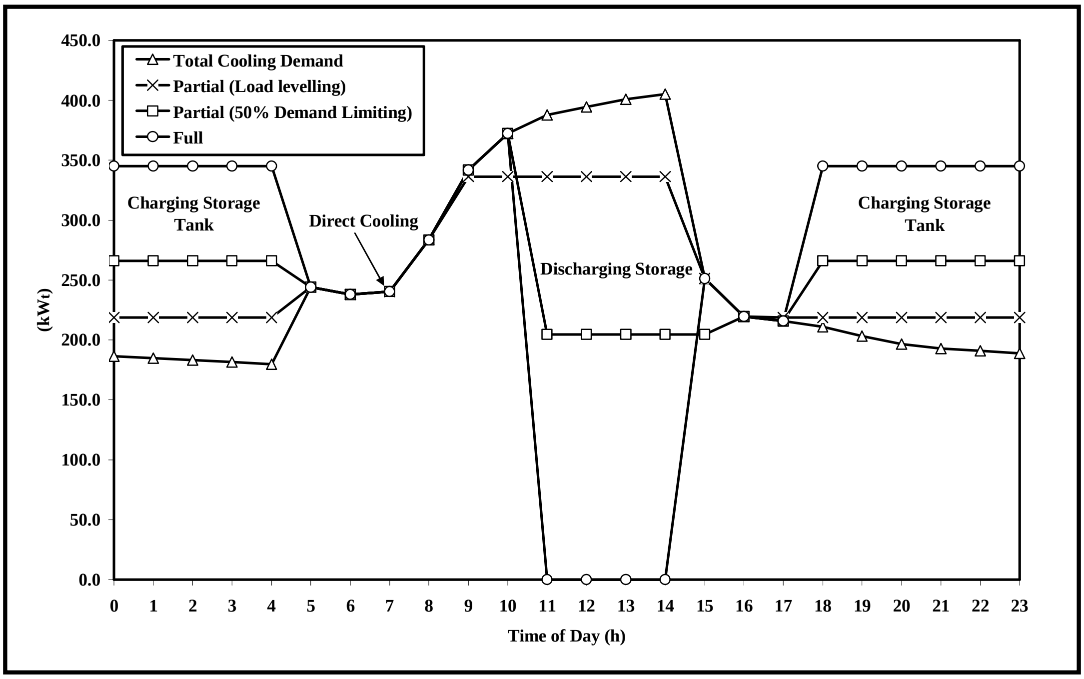

Figure 5.4 Hourly profile of cooling production of the chiller in ice storage AC systems.

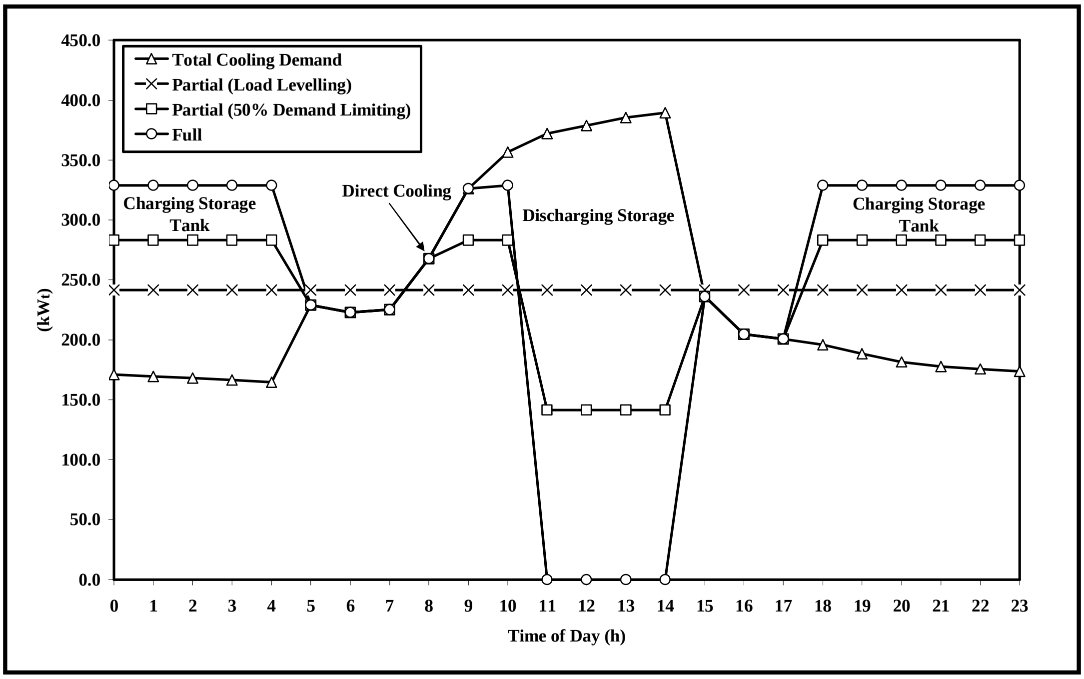

Figure 5.5 Hourly profile of cooling production of the chiller in chilled water storage AC systems.

Figures 5.4 and 5.5, show the cooling production by the chiller when the ice of a chilled water storage AC system is designed with a load levelling operation that operates at full capacity as much as possible during the day. When the chiller capacity is higher than the system load, the excess cooling from the chiller is stored in the tank. When the chiller capacity is less than the cooling load, the additional requirement of cooling is provided from the tank. Furthermore, with a 50% demand limiting operation strategy, the chiller operates at full load during the charging hours to meet directly the cooling load of the building and simultaneously charge the tank. During the discharging hours (for example, from 11:00 hours until 15:00 hours), the chiller operates at a reduced capacity of 50%. Similarly, in full storage operation strategy, the chiller operates at full load during the charging hours (for example, night time), and during discharging hours (for example, day time) the chiller switches off completely and the cooling load is met directly by the tank.

A comparison of Figures 5.4 and 5.5 shows that the difference in the cooling productions of the chillers during the charging hours is small operating with ice storage and chilled water storage systems. Because the capacity ratios CR_{char} of the chillers in ice storage systems are lower than those in chilled water storage systems, and because the peak and the total integrated cooling loads of ice storage AC systems are higher than those of chilled water AC systems, the sizes of the chillers in ice storage AC systems are larger than those in chilled water storage AC systems when the same operation strategies are implemented (see also Table 5.1).

The storage capacity for each operation strategy in Figures 5.4 and 5.5 is equal to the area between the cooling load and cooling production during the charging hours. It can be observed from the two figures that the storage capacities of the load levelling and 50% demand limiting operation strategies in chilled water systems are higher than in ice storage systems. This is because the chillers in chilled water AC systems are smaller than are those in ice storage AC systems; hence, more cooling is taken from the storage tank. In a full storage operation strategy, the difference in the storage capacity between ice and chilled water storage AC systems is small (see Table 5.2).

5.5.4 Air handling units

Generally, the size of the AHU depends on the design of the volumetric flow rate of the moist air and on the capacity of the cooling coil within the AHU. Furthermore, other factors that have a significant effect on the size of the AHU, particularly on the pressure drop within the AHU, are the mixing box, filters, heating coils and so on; as the pressure drop increases, so the size of the fan and motor will also increase and, hence, so will the capacity of the cooling coil. The capacity of the cooling coil must be able to meet the highest sum of the instantaneous space loads for all the spaces served by the coil, plus any external loads such as fan heat gain, duct heat gain, duct air leakage, and outdoor air ventilation loads (sensible and latent).

To select correctly the AHUs for conventional, ice, and chilled water storage AC systems, the volumetric flow rate of air for each zone or for the AHU was calculated based on the design sensible load; the psychrometric properties of the moist air at different states were determined in the air distribution system as discussed in Appendix A.

The states of the air distribution system are shown in Figure 5.6, and denoted as point R, which represents a state point in the return air from the zone; S, which represents the state of the supply air to the zone; M, which represents the off-coil mixing condition; and C, which represents the on-coil state. Other state points in which the psychrometric properties of the air were determined are point Z, which represents the condition at the zone in the building, and point O, which is the outside design condition.

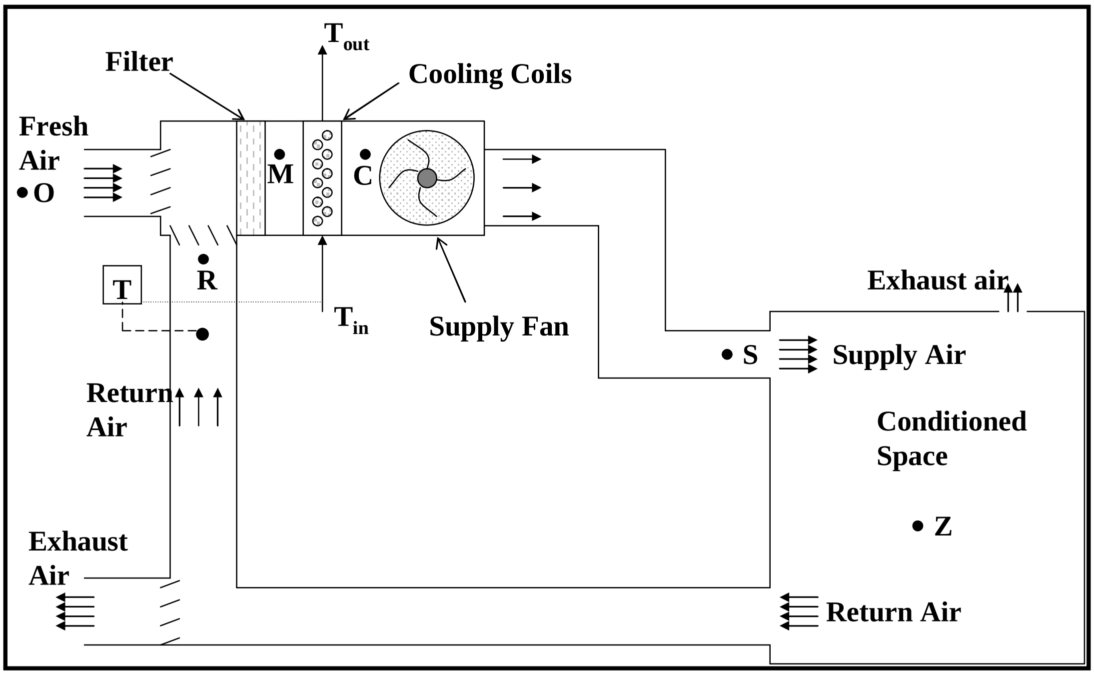

Figure 5.6 The states in the air distribution system.

The dry and wet bulb temperatures, the humidity, and the enthalpies at different states in the air distribution systems were calculated using the formulae for calculating the psychrometric properties of moist air as given in the ASHRAE handbook (Jones, 1994; ASHRAE, 2001d) and shown in Appendix A. The heat gains by the supply and return ducts and the fan motors as well as the ventilation load were estimated. Finally, the capacity of the cooling coil for each AHU was obtained.

Based on the determination of the capacity of the cooling coil, on the volumetric air flow at the design off-coil air temperature and humidity, which were determined to meet each zone’s temperature requirement, design air-side and water-side entering and leaving temperatures, and on the chilled water flow rate and pressure drops, the FLEX-air Version 5.08SA software program (Carrier, 2005) was used to size each AHU. For each AHU, the pressure drop within the AHU, the off-coil dry bulb temperature, and the humidity and the motor and fan efficiencies were obtained (for further details of specifications refer to Appendix B).

5.5.5 Pumps

Liquid pumps used in the AC systems industry to circulate chilled water or brine are usually of the centrifugal type. To size properly and hence select a centrifugal pump for a given application, two important parameters must be obtained: the total pumping head and the design volumetric water flow rate. The total pumping head is the sum of the static head and the friction head.

To determine the total pumping head, the chilled water piping must be first designed and then the volumetric flow rate must be then calculated. For conventional, ice storage, and chilled water storage AC systems, the piping distribution systems were arranged with primary-secondary piping designs (James, 1996; ASHRAE, 2000c). The layout and arrangement of the piping systems are shown in Appendix D. The design volumetric flow rates of the chilled water were calculated in the primary and secondary piping circuits and are summarised in Table 5.5 for all AC systems.

Air Conditioning System	Volumetric Design Flow Rate (m^3s^{-1})	
	Primary Circuit	Secondary Circuit
Conventional	0.0165	0.0169
Ice Storage (partial Load Levelling)	0.0153	0.0114
Ice Storage (partial 50% Demand Limiting)	0.0178	0.0114
Ice Storage (Full)	0.0253	0.0114
Water Storage (partial Load Levelling)	0.054	0.0100
Water Storage (partial 50% Demand Limiting)	0.074	0.0100
Water Storage (Full)	0.080	0.0100

Table 5.5 The calculated volumetric design flow rate in the primary and secondary circuits

The volumetric design flow rate for each AC system in the primary circuit is equal to the design chilled water flow rate through the chiller, which was obtained from the selection program ECAT2 Version 4.12 (Carrier, 2004). The volumetric design flow rate for each system in the secondary circuit was computed based on the design cooling coil capacity and the design of the temperature differential across the coil (see Appendix D for further discussion). By using the above volumetric design flow rates, the pipes were sized based on a recommended volumetric flow rate range for which the pressure drop within the pipe does not exceed 391 Pam^{-1} and velocity does not exceed 3.1 ms^{-1} for closed piping system (El-Amer, 2005).

Having determined the flow rate and obtained the sizes of the pipes, the friction factor was calculated using the Colebrook equation to determine the friction losses of the pipes. Furthermore, the friction losses of the valves then were computed based on the data available in several studies for example, (Carrier, 1965; Crane, 1988) for calculating the resistance coefficient for each type of valve in the systems. The number and type of valve associated with different components in an AC system are shown in Appendix E (KEO, 2004). The result of these calculations is shown in Table 5.6.

Head Loss	Conventional		Ice Storage System						Chilled Water Storage System					
			Partial				Full		Partial				Full	
			Load Levelling		50% Demand Limiting				Load Levelling		50% Demand Limiting			
	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.
Pipes (m)	2.1	8.0	2.7	4.4	1.4	4.4	2.6	4.4	2.5	4.2	3.8	3.6	2.6	4.0
Valves and Fittings (m)	6.7	6.8	13.7	10.0	7.9	10.0	13.6	10.0	5.4	6.2	9.4	4.5	7.6	7.1
Coil (m)	0.0	1.0	0.0	3.5	0.0	3.5	0.0	3.5	0.0	4.3	0.0	4.3	0.0	4.3
Chiller (m)	2.5	0.0	4.2	0.0	3.4	0.0	4.2	0.0	0.9	0.0	0.9	0.0	1.1	0.0
Storage Tank (m)	0.0	0.0	11.5	0.0	11.5	0.0	11.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Total (m)	11.3	15.8	32.1	17.9	24.3	17.9	32.0	17.9	8.8	14.6	14.2	12.4	11.3	15.4

Table 5.6 Calculated pumping head in the primary and secondary circuits

Table 5.6 contains the required pumping head (for example, friction losses) through the various system components, such as pipes, fittings, chillers, ice tanks, and cooling coils. The friction losses of the diffuser pipes and valves that were designed with the chilled water tanks were included with the pipes and fitting losses. From Table 5.6, it can be seen that the total friction losses in the primary circuit of ice storage AC systems are very large because of the large pressure drop in the ice tank. Moreover, the friction losses in the chilled water storage systems (except in the primary circuit with 50% demand limiting partial storage) are lower than in the conventional AC system because of the small friction loss in the chiller and diffuser pipes in the chilled water tank.

Using the calculated volumetric design flow rate and friction losses, the primary and secondary pumps were selected and their specifications were obtained from the pumps' suppliers. For proper pump selection, other information such as type of liquid, operating temperature range locations, and so on, was given to the supplier to select the correct pump (refer to Appendix B for pump specifications).

5.6 Auxiliary systems heat gain

In Section 5.5, the sizing procedure of the components in the AC systems was discussed and the specification was obtained for each of these components. This section discusses the calculations of the heat gain by the auxiliary systems including AHU fan and pump motors based on the obtained specifications as shown in Appendix B as well as the heat gains by the ducts and pipes.

5.6.1 Air handling unit fan motors

The supply and return air fans and their electric driven motors in the AHUs dissipate heat to the air stream. The amount of heat dissipation depends on the static pressure of the duct system, and the efficiency and locations of the fans and motors. The heat gain by the inefficiency of the motor is significant and may reach as high as 15% of the system peak cooling load; therefore, it must be calculated and added to the building cooling load. In calculating the heat gain by the motors and fans the following points must be considered:

a. Efficiency varies with the type and size of motor; for initial estimation, the AC designer can refer to the ASHRAE handbook (ASHRAE, 2001c). The values of the motor efficiency reported in the handbook gave average efficiencies and related data regarding typical electric motors, and were derived from lower efficiencies reported by several manufacturers. For some approximation

i. Motors of range 0.037 to 0.56 kW_e, efficiency ranges from 35% to 72%

ii. Motors of 0.75 to 3.73 kW_e, efficiency ranges from 75% to 82%.

iii. Motors of 7.47 kW_e and above, efficiency ranges from 85% to 91%.

b. If the motor is inside the air stream, but the driven fan is outside, only the heat gain by the inefficiency of the motor is counted.

c. If the motor is outside the air stream, and the driven fan is inside, only the heat gain by the fan inefficiency is counted.

d. If both the motor and fan are inside the air stream, their entire energy will be dissipated into the air stream, and the total heat gain by both of motor and fan inefficiencies is counted.

Since the motors and fans are within the air stream in the AHU, therefore the heat gain by the motor and fan inefficiencies was calculated (ASHRAE, 2001c) by

$$\dot{Q}_m = \frac{\Delta p_a \dot{V}_a F_{um} F_{lm}}{E_m E_f} \tag{5.5}$$

In equation (5.5), the product of the fan total pressure Δp_a and the volumetric flow rate of the supply air $\dot{V}_a$ are known as air power. Furthermore, the motor rated power is equal to the air power divided by the fan total efficiency E_f. The fan total pressure Δp_a in equation (5.5) is equal to the summation of the external pressure drop (for example, pressure in the duct) and the pressure drop within the AHUs. The pressure drop in the ducting system depends on the velocity of the air distribution in the duct and the complexity of the ducting systems, and was assumed to be 398 Pa and to be the same for all AC systems. The coil, filters, and mixing box are the main contributors to the pressure drop in the AHUs and their values obtained from the AHU selection software (Carrier, 2005).

The volumetric flow rate of the air $\dot{V}_a$ was calculated as shown in Appendix A. Moreover, for all AHUs, the motors' efficiencies E_m were obtained from their catalogues (BROOK CROMPTON, 2005) and the fans' total efficiencies E_f were obtained from the AHU selection software (Carrier, 2005).

The AHUs are operating continuously for 24 hours a day throughout the year, therefore, the motors use factor F_{um} the values for which were set to be equal to 1. Furthermore, the motor load factor F_{lm} is defined as the rated load delivered under the conditions of the cooling load estimate (McQuiston, 1992). Generally, the heat output from the motor is proportional to the motor load within the overload limit. Because of the typically high no load motor current and fixed losses, the value of the motor load factor F_{lm} is generally assumed to be equal to 1 (McQuiston, 1992); hence, the motor load factors F_{lm} were set equal to 1 in the equation.

5.6.2 Pump motors

The main function of the pumps in a central AC system is to transport the chilled water from the chillers located in the central plant to the various AHUs in the building and then back from the AHUs to the chillers through a designed piping network.

The power input in kilowatts to the pumps to drive the chilled water adds heat to the system. The amount of additional heat depends on the pump capacity, total pumping head, the efficiency of the pumps and motor and the locations of the pumps and motors. Equation (5.5), which was used to calculate the equivalent heat gain $\dot{Q}_m$ by the motors of the fans in the AHUs was also used to determine the heat gain by the motors of the pumps $\dot{Q}_p$. The volumetric flow rate of air $\dot{V}_a$, was replaced by the volumetric flow rate of the chilled water $\dot{V}_w$; fan efficiency E_f was replaced by pump efficiency E_p; and the total pressure drop of the fan Δp_a was replaced by total pump pressure drop Δp_w of the chilled water in the piping circuits. Therefore, the equivalent heat gain of the pumps' motors was computed by

$$\dot{Q}_p = \frac{\Delta p_w \dot{V}_p F_{um} F_{lm}}{E_m E_p} \qquad (5.6)$$

Equation (5.6) was used to calculate the heat gain by both the primary and secondary pumps. The heat gain $\dot{Q}_p$ depends on the number of pumps in operation. In equation (5.6), the motor and pump efficiencies for both the primary and secondary pumps were obtained from the pump supplier and are listed in Appendix B. The use F_{um} and load F_{lm} factors of the motors were both assumed to be equal to one.

The pressure drops (that is, total pumping head) Δp_w and the volumetric design flow rate $\dot{V}_w$ (that is, pump capacity) have already been determined for the primary and secondary pumps (refer to Tables 5.5 and 5.6). The chiller pump (or primary pump) was assumed to be running independent of the building load except for in a chilled water storage AC system, which is designed for full storage operating strategy whereby the primary pump is switched off completely. In the secondary distribution pumps, the number of pumps in operation depends on the variation of the load on the coil in the AHUs, and therefore it is a function of the volumetric flow rate that is supplied to the cooling coil to cope with the load.

For some applications of using ice thermal storage systems where the percentage of the glycol solution is high and its operating temperature is very low, the viscosity of the fluid will be much higher than the viscosity of water, therefore a correction factor for the pressure drop Δp_w volumetric flow rate $\dot{V}_w$ and pump efficiency E_p must be applied to equation (5.6). However, for the application of the ice thermal storage studied in this report, these corrections were ignored because they did not have a significant effect on the pumps because of their small percentage of solution and temperature range (Rishel, 1997).

5.6.3 Ducts

Duct heat gain is the heat transfer caused by the temperature difference between the ambient air surrounding the duct and the air flowing inside the duct. The heat gain of the duct is one of the parameters that affect the supply and return air temperatures as well as the supply and return air flow rate in the ducting system. If the supply duct is located in a hot roof space exposed directly to the sun and not insulated, the heat gain may rise to as much as 25% of the total sensible load (Norman, 1983). Consequently, the duct should be well insulated to minimise these heat gains. Duct insulation is also necessary even when the duct is located in an air conditioned space to prevent the condensation of moisture on the cold surface of the supply duct carrying the cold air.

The heat gain must be known for the calculation of the supply air quantities, supply air temperatures, and coil loads. ASHRAE (2001a) provides details of the estimation of the heat gain from the duct if the duct dimensions and inlet, exit, and surrounding temperatures of the duct and average velocity of the air are known. However, if such data are not available, the rise in the temperature of the duct can be assumed, which ranges between 0.1° C to 2° C for the ducts of the supply air and from 0.2° C to 1.3° C for return or recirculated air ducts (Jones, 1994). Based on these assumptions, the volumetric flow rate and temperature of the supply air and the load on the cooling coil can be determined using a psychrometric analysis of the AC system.

Furthermore, a rough estimate of the temperature increase of the supply air per unit duct length for an insulated duct is reported by (Shan, 2001) for two supply air velocity ranges. For supply air velocity of less than 10 ms^{-1}, the air temperature rise is 0.0185° Cm^{-1}, and for a velocity greater than 10 ms^{-1}, the rise of the air temperature is 0.0139° Cm^{-1}.

In Kuwait, most of the AC designers avoid locating the supply and return ducts outside the building because of the high ambient temperature. As a result, the temperature rises in the ducting system are always small and reach a maximum of about 0.2° C (El-Amer, 2005). The heat gains by the supply and return ducts were computed based on a temperature rise of 0.15˚ C using the following

$$\dot{Q}_d = \dot{V}_a \rho_a c_p \Delta t_d \tag{5.7}$$

The volumetric flow rate of the moist air in equation (5.7) was obtained using the sensible heat gain for each zone in the building as given in Appendix A.

5.6.4 Pipes

Pipes carrying chilled water should be thermally insulated to reduce the heat gain of the chilled water and improve the energy efficiency of the system. The chilled water pipes are usually exposed to the sun and subjected to a high ambient temperature. All pipes, therefore, must be insulated; the optimum thickness of the insulation of the pipes depends mainly on the operating temperature of the inside flowing water, the pipe diameter, and the type of service. ASHRAE/IESNA Standard 90.1-1999 specifies the minimum pipe insulation thickness for water systems and these are listed by Shan (Shan, 2001). It is recommended in this standard that pipes of 50 mm to 150 mm diameter with a fluid design operating temperature range of 22.2° C to 33.3° C should have a minimum pipe insulation thickness of 25 mm.

The operating temperature range of the chilled water AC systems in the Kuwaiti climate is higher than the one given in the ASHRAE Standard; it may reach about 45° C for conventional and chilled water storage systems, and about 53° C for ice thermal storage systems. Therefore, the guideline for the minimum insulation thickness that is provided by the MOE was used. Pipes of size 50 mm diameter must be insulated with a minimum insulation thickness of 25 mm, and pipes of larger sizes, from 60 mm to 150 mm diameter, must be insulated with at least 50 mm insulation thickness. Moreover, the guide recommends that the pipes must be insulated with rigid fibreglass of not less than 96 kgm^{-3} density or equivalent.

The heat transfer rate to the chilled water in the pipes was calculated using a one dimensional heat equation for a composite hollow cylinder, that is,

$$q = UA\left(T_a - T_w\right) \tag{5.8}$$

The overall heat transfer coefficient U for insulated by was obtained by

$$U = \frac{1}{\frac{1}{h_w} + \frac{r_1}{k_p}\ln\left(\frac{r_2}{r_1}\right) + \frac{r_1}{k_i}\ln\left(\frac{r_3}{r_2}\right) + \frac{r_1}{r_3}\frac{1}{h_a}} \tag{5.9}$$

The internal pipe area was determined using the following equation

$$A = 2\pi r_1 L \tag{5.10}$$

Equations (5.8) to (5.10) were used to calculate the heat gain by the chilled water in the piping systems, and the pipe's internal and external radii were obtained from the ASHRAE handbook (ASHRAE, 2000e); for schedule 40 steel pipes, the thermal conductivity of the steel pipe is 74 $Wm^{-1}{}^{o}K^{-1}$, and that for the insulation (fibreglass with 96 kgm^{-3} density) material is 0.035 $Wm^{-1}{}^{o}K^{-1}$ (KIMMCO, 2005). Moreover, the convective heat transfer coefficients h_w and h_a were neglected since the effect of the convection heat transfer is very small compared to that of the conduction.

5.7 Final cooling load profiles

Having calculated the heat gain by the auxiliary systems, the system load profiles for the conventional, ice storage, and chilled water storage AC systems were obtained by adding the building cooling load and the calculated heat gains by the auxiliary systems. The results of the calculations were compared with previously assumed heat gains (see Section 5.2). Table 5.7 provides a summary of the comparisons for the maximum and the total integrated cooling load for all AC systems.

Air Conditioning System	Cooling load			
	Maximum (kW_t)		Total Integrated (kW_th)	
	Calculated	Assumed	Calculated	Assumed
Conventional	371.6	393.4	5107.6	5624.5
Ice Storage (Partial Load Levelling)	376.3	405.2	5226.7	5906.6
Ice Storage (Partial 50% Demand Limiting)	375.5	405.2	5207.3	5906.6
Ice Storage (Full)	381.1	405.2	5334.8	5906.6
Water Storage (Partial Load Levelling)	367.6	389.6	5027.2	5556.4
Water Storage (Partial 50% Demand Limiting)	368.2	389.6	5043.8	5556.4
Water Storage (Full)	366.9	389.6	5041.4	5556.4

Table 5.7 Result summary of the calculated maximum and integrated system load

Since the calculated maximum cooling load and the total integrated cooling load are both higher than the assumed values, therefore, the calculation of the chiller and storage sizes and heat gain need not be repeated. The final profiles are therefore generated for all AC systems and are shown in Figure 5.7.

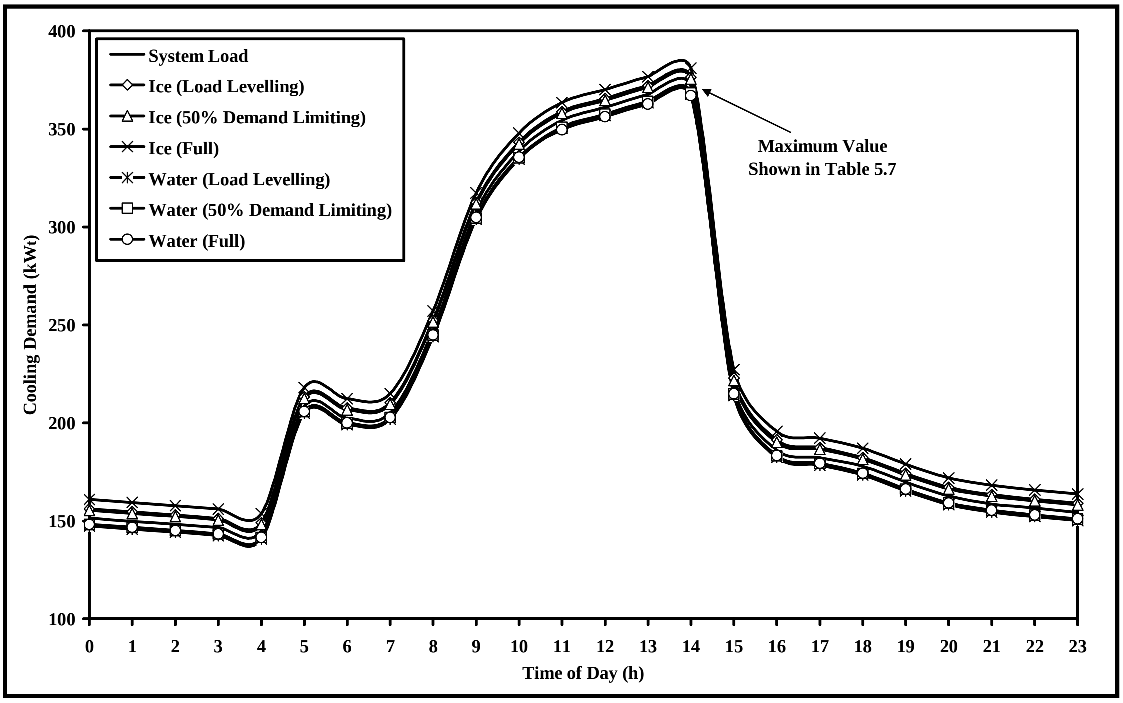

Figure 5.7 Hourly system load profiles of the AC systems.

From Figure 5.7, it can be seen that there are small differences in the system load profiles, because the building cooling load is the same for all the AC systems; therefore, the difference in the profiles stems mainly from the differences in the heat gain by the auxiliary systems. In ice storage AC systems, the auxiliary system heat gains resulting from the AHU fan motors, secondary pump motors and ducts are the same; however, the heat gains by the primary pump and by the pipes are slightly different. The heat gain by the primary pump in the full storage operation strategy is slightly higher because of the bigger pump size compared to other operation strategies. In fact, the ice storage system operating with full storage operation strategy has the highest maximum and total integrated cooling loads compared to other AC systems.

Moreover, it was found that the calculated heat gains by the auxiliary systems in all operation strategies in chilled water storage AC systems are lower than those in the conventional and ice storage AC systems. The profiles that are shown in Figure 5.7 represent the final load shape of each design of the AC system considered in this report. Since the chillers that are selected for each AC system must be able to cool the building as well as counteract the heat gain by the auxiliary systems, their power and energy consumption strongly depend on the profiles of Figure 5.7, and this will be discussed in more detail in Chapter 6.

5.8 Conclusions

In this chapter, the sizing of each component in the conventional, ice storage, and chilled water storage AC systems was obtained. The chillers and storage sizes were first estimated using the system load profiles of the building, which were based on an assumed percentage of heat gain by the auxiliary systems. The chiller and storage sizes were calculated based on the requirement that the total integrated capacity of the chiller must be equal to the integrated system load. For 50% demand limiting partial and full storage operating strategies charging was selected to be between 18:00 hours to 5:00 hours and discharging to be between 11:00 hours to 15:00 hours.

The sizes of the AHUs in each AC system were obtained based on the conducted analysis of the psychrometric properties of moist air at different states in the air distribution system. Furthermore, the sizes of the primary and secondary chilled water pumps were computed using the calculated design volumetric flow rates of the chilled water. The total pumping head was calculated in both the primary and secondary circuits resulting from the pressure drop in pipes, valves, and various heat exchangers in the piping circuits.

Based on the above selected components, the heat gains by the auxiliary systems including AHU motors, pump motors, supply and return ducts and chilled water pipes were properly recomputed, and added to the building cooling load, which was obtained from the ESP-r building simulation (see Chapter 4). This chapter was concluded with properly defined system load profiles for a typical medium size building such as the clinic under study. The developed profiles will then be used to examine the energy performance of the AC systems as will be shown in Chapter 6.

Based on the sizing procedure discussed in this chapter, the components in the AC systems were selected. These will help to estimate the capital and maintenance costs as will be further illustrated in Chapter 7 where the life cycle costs and the economic impact of ice and chilled water storage on the government and users will be assessed.

Chapter 6

Energy Performance Analysis

6.1 Introduction

In Chapter 5, the sizing of each component in conventional, ice and chilled water storage AC systems was determined. The heat gains by these components were calculated and added to the building cooling load of the clinic building where the system load profiles were developed.

In this chapter, one of the objectives of this research work is presented, the estimation and comparison of power and energy consumptions of conventional, ice and chilled water storage AC systems. As part of the work, the power consumption at the peak cooling load and energy consumptions at design condition, and annual energy consumption of ice and chilled water storage AC systems relative to the conventional system is made.

In order to compare and analyse the energy performances of AC systems, the input power requirements of the components including chillers, AHUs and pumps are determined for the design day condition and for the whole year. The power and energy consumptions of the chillers are a function of system load and dry bulb temperature and can be estimated using their available performance data. The power consumptions of the AHUs and pumps depend on the sizes of their motors which are given in their specification tables in Appendix B.

For the design day condition, first, the proportion of the power and energy consumptions used by the components in each AC system is discussed, second, the change of the power at maximum cooling load and energy consumptions of ice and chilled water systems relative to conventional system are presented.

The energy consumption for each system for the whole year is also estimated and analysed. The proportion each component uses of the annual energy consumption is established. The change in the energy consumption each component in the ice and the chilled water storage system is compared with the conventional AC system.

6.2 Input power requirements

In this section, the input power requirements are calculated for each component including chillers, AHUs, and pumps in the AC systems. The input power of the chillers depends on numerous factors, such as the cooling load, condenser temperature (a function of outside weather conditions), and the control strategies.

Moreover, the motor input power on the AHU depends on the fan and motor efficiencies, the pressure drop in the duct and within the AHU, and on the volumetric flow rate of the design supply air. Similarly, the motor input power of the pump mainly depends on the motor and pump efficiencies and on the calculated design volumetric flow rate and pumping head.

6.2.1 Chillers

ECAT2 (Carrier, 2004) provides performance data of the chillers based upon a factory run test for each model of the chiller at the full load design point and at the part load operating points as required. The performance data of the selected chiller at part load operating points was obtained, including percentage load, gross capacity, absorbed power and COP. This was achieved by manually entering the percentage unloading steps and the dry bulb temperatures of the condenser inlet air. In order to obtain a proper step in the percentage of the gross capacity (for example, a 5% step in gross capacity and a 3° C step in dry bulb temperature were used in this report: see the performance data of the chillers in Appendix F) the input of the unloading steps was adjusted accordingly.

For the conventional chiller, the capacity and the power consumption of the chiller were obtained based on the calculated hourly system load and the hourly design dry bulb temperature. At each hour, the capacity ratio of the chiller was calculated at the given dry bulb temperature of the design day and load. The capacity ratio of a chiller is defined as the chiller capacity at a given dry bulb temperature normalised by the capacity of the same chiller when operating at full load at the same dry bulb temperature.

In the case of chillers running with ice and chilled water storage AC systems, it was assumed that the chillers were operating at full capacity during the charging period (that is, 100% capacity) and during the discharge period were either switched off, operating at full load, or operating at a given percentage capacity (for example, 50% demand limiting) depending on the operation strategy. During the charging time, it was assumed that the chillers would match the system load, at the same time charging the ice tank or storing cold water in the storage tank. When the ice or water tank is fully charged, it was assumed that the chillers would directly match the system load.

Having obtained the performance data of the chiller at different proportions of gross capacity and inlet air design dry bulb temperatures for each operating strategy, the cooling production and power consumption required by the chiller for the clinic building were determined by a simple two dimensional interpolation of the data obtained for each chiller model.

6.2.2 Air handling units

The calculated heat gain in kilowatt cooling by the AHU motors using equation (5.5) (see Chapter 5) is simply equal to the input power in kilowatt electricity to the motors of the AHU. Therefore, equation (5.5) was also used to calculate the input power requirements for all motors of the AHU in the building. Furthermore, it was assumed that the motors would run continuously throughout the year irrespective of the cooling load of the building; thereby, the motors would run at a constant speed. The input power of each motor is listed in Appendix B and was obtained from the FLEX-air Version 5.08SA software program (Carrier, 2005); the values were based on the calculated pressure drop and design volumetric flow rate.

6.2.3 Pumps

Similar to the AHU motors, the input power of the primary and secondary pump motors was calculated using equation (5.6) (see Chapter 5). For the primary chilled water pumps (for example, chiller), the input power of the motors was calculated based on the assumption that the pumps would run continuously when the chillers were operating. For the secondary chilled water pumps, the input power is determined based on the number of pumps in operation. The number of pumps that come into operation is a function of the volumetric flow rate of chilled water through the cooling coil, which varies depending on the cooling load of the building. Further details of the pumping head, flow rate, and motor and pump efficiencies for the primary and secondary pumps, which were used in the calculation of the power input, are given in Appendix B.

6.3 Chiller demand profiles

In this section, the profiles of the cooling production, power consumption and the COP of the chillers in the conventional, ice, and chilled water storage AC systems are discussed and analysed in detail, since the use of the storage systems has a significant impact on the chillers' performance compared to that of other components in the AC systems, such as AHUs and pumps.

As is mentioned elsewhere in this report, the AHUs operate 24 hours a day throughout the year irrespective of the building's cooling load; therefore, their power demand profiles are constant throughout the day. Likewise, the power demand profiles of the primary pumps are similar to those of the AHUs; however, in the case of a chilled water storage AC system operating in full strategy, the primary pumps are entirely switched off during the discharging time (for example, from 11:00 hours to 15:00 hours).

For the secondary pumps, the power profiles are similar for all AC systems; however, the power requirement is different due to the differences in their sizes. From the numerical data obtained for the secondary pumps, it was observed that during the night time, when the cooling load is low, two of the secondary pumps would need to be in operation, and during the day time when the cooling load is high three pumps would need to be in operation.

In the following subsections, the profiles of the cooling production, power, and coefficient of performance of the chillers operating in conventional, ice, and chilled water storage AC systems are examined. These profiles for the chillers were obtained using the design day dry bulb temperatures of Kuwait given by Shaban (2000) and the developed system load profiles as shown in Figure 5.7.

6.3.1 Cooling production

Figure 6.1 illustrates the hourly cooling production profile for the conventional chiller and for chillers operating with ice and chilled water storage AC systems for different operating strategies, load levelling, demand limiting partial storage and full storage. In the conventional AC system, the chiller operates continuously throughout a 24 hour period to provide sufficient cooling to the building as shown in the figure. From Figure 6.1, it can also be observed that the profile of the cooling production is similar to the profile of the system load profile shown in Figure 5.7 for the conventional system. In this configuration, the chiller runs for most of the time at part load and at full or nearly full load for only a few hours during the maximum cooling load period.

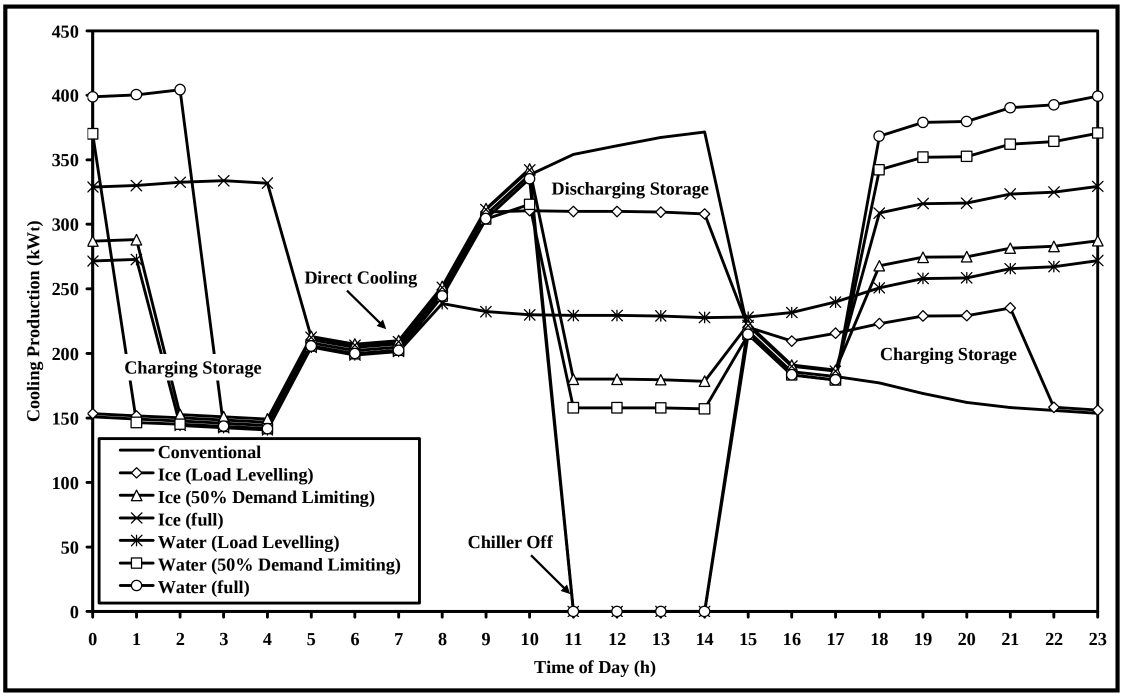

Figure 6.1 Hourly cooling productions profiles of the chillers at design condition.

For the ice storage AC system operating with a load levelling strategy, the chiller starts charging the storage at 15:00 hours when the chiller cooling production in charging mode is higher than the cooling load, and ends at 22:00 hours; so, the chiller takes approximately seven hours to charge the storage tank. At 9:00 hours the cooling load exceeds the chiller capacity, the excess cooling load is met by the ice tank until 15:00 hours where the cooling load is met by only the chiller.

Comparing Figure 6.1 and Figure 5.4 where the chiller is initially selected as discussed in Section 5.5.1, it can be seen that the chiller in Figure 6.1 took less time to charge the ice tank although the chiller that was selected using the selection software was smaller than the calculated chiller obtained from equation (5.1) (refer to Figure 5.4 and also see Table 5.1). This is because the selected nominal chiller capacity using the software was selected based on an assumed system load (for example, see Figure 5.1 and Table 5.7) where the maximum and the total integrated cooling load are both higher than the calculated one. Furthermore, the capacity ratio of the chiller during the charging mode was found to be about 0.67, which is higher than the 0.65 that was assumed for the initial chiller sizing. In addition, during night time operation the chiller produces more cooling as shown in Figure 6.1, because of the lower dry bulb temperature at night.

In the same way, the chiller in the chilled water storage AC system with load levelling operation strategy takes less time to charge the storage tank. The chiller starts to charge the storage tank at about 15:00 hours when the capacity of the chiller is higher than the cooling load, and ends at about 2:00 hours when the tank is fully charged.

In a 50% demand limiting partial storage operation strategy, it can be observed from Figure 6.1 that the chiller capacities for both chillers in the ice and chilled water storage systems are reduced to 50% to cope with system load during the discharging time from 11:00 hours to 15:00 hours, and the extra cooling requirement is met by the stored ice or water in tank. In practice, this is achieved for the ice storage AC system by increasing the chiller outlet fluid temperature, or by unloading the chiller to a given predetermined percentage of the load. However, for the chilled water storage AC system, this can be achieved by passing 50% of the chilled water produced by the chiller through the secondary circuit and back to the chiller again using a two or three way valve.

During the night time, both chillers in the ice and chilled water storage AC systems start charging the tank at about 18:00 hours as designed (see Section 5.4); the chiller in the ice storage systems ends at about 2:00 hours, approximately one hour earlier than the chiller in the chilled water storage system. Generally, the demand limiting operation strategy represents a compromise between full and load levelling partial storage. In addition, the ability to shift the time of the maximum power demand is higher than for the load levelling system and lower than for the full storage system. However, with this operating strategy, the control system is complicated since the maximum demand must be met through the storage. Typically, an electric demand meter is used for this purpose (Hasnain, 1998).

Moreover, when the chiller operates at full load during the night time, more cooling is produced because of the lower dry bulb temperatures at night. This system design is effective where the maximum cooling load is much higher than the average cooling load and also expends the lowest initial cost compared to conventional and other storage strategies (Dincer, 2002a).

The chillers in the full storage system switch off during the discharging time from 11:00 hours to 15:00 hours as illustrated in Figure 6.1 at which time the cooling load is directly met by the storage tank. During the charging time, from 18:00 hours to the time when the storage tank is fully charged, the chiller operates at full capacity to produce the required cooling for the building and to charge the storage tank. Dincer (2002a) argues that the full storage system is likely to be attractive if high peak demand charges apply, if there are short overlaps between peak load and peak energy periods, or if daytime energy rates are based on short duration peak periods.

6.3.2 Power consumption

It is very important to look at the power consumption profiles of the chillers for the AC systems during the 24 hour operation, since the main purpose of using the cool storage technologies is to reduce the power consumption in the day time; this can be seen from Figure 6.2 below.

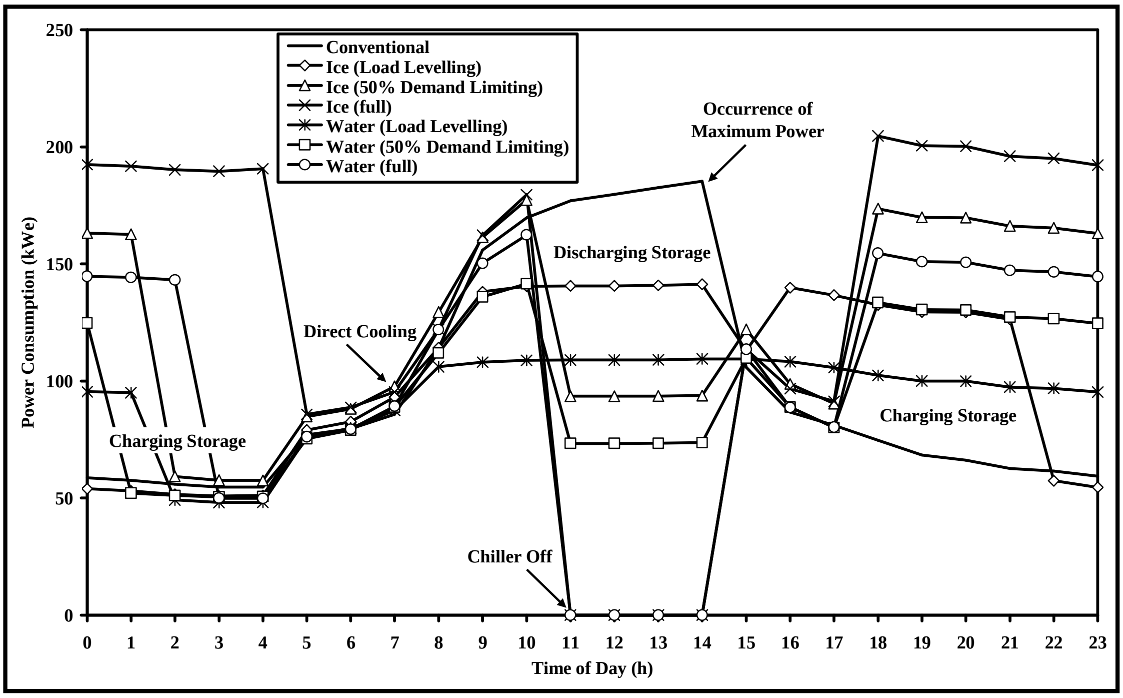

Figure 6.2 Hourly power consumptions profiles of the chillers at design condition.

Figure 6.2 shows a plot of the hourly power consumption profiles of the chillers for conventional ice and chilled water storage AC systems. Generally, the power requirement of the chillers increases with the cooling load of the building (that is, load on the chiller). The maximum power demand occurs in the day time particularly when the cooling load is at a maximum. The power consumption of the chiller in the conventional system directly increases with the system load, and is highest during the day time from 10:00 hours to a maximum at 14:00 hours and decreases during the night time when the cooling load is lower. Therefore, it can be seen from the figure that the power consumption by the chiller in the conventional system is much higher during the day time compared with the ice and chilled water storage systems.

For chillers operating with ice and chilled water systems, except load levelling storage, the shapes of the power profiles as shown in Figure 6.2 have an opposite trend compared to the chiller in the conventional AC system. During the night time, the chillers produce more cooling and hence consume more power than during the day time. This is because at night time, the chillers operate at full load to cool the building and to charge the ice or chilled water storage tank. Furthermore, since in the day time between 11:00 hours to 15:00 hours, the cooling load is met fully or partially by the storage tank, the load on the chillers decreases, hence, the power consumptions as shown in the figure. The degree of power reduction depends on the design operating strategy of the storage AC system and chiller size.

The power profiles of the chillers of the load levelling operation strategy in both ice and chilled water storage systems are nearly constant during the 24 hour period except for a few hours late at night and early in the morning, from 22:00 hours to 6:00 hours in the case of ice storage systems, and from 3:00 hours to 6:00 hours in the case of chilled water storage systems, when the storage tank is fully charged and the cooling load is directly met by the chillers, thus further reducing the power consumption.

In this operation strategy, the chiller runs continuously at full load hence consuming more power to charge the storage tank at night and to meet the building load during the day. From Figure 6.2, it is clear that the power consumption of the chillers decreases slightly during the night time operation, from 18:00 hours to 0:00 hours, indicating that the chillers operate more efficiently and with a higher coefficient of performance during the night because of the reduced dry bulb temperature. It has been reported in (Hasnain, 1998) that in this operation strategy a reduction of 40 to 60% of the peak electrical demand for cooling may be achieved during the discharging time.

Moreover, from the plot of the hourly power consumption profiles in Figure 6.2, it can be observed that the power consumption of the chillers operating with a 50% demand limiting strategy during the discharging time is lower than that of the chillers operating with a load levelling operation strategy and higher than that of the full storage strategy and this occurs for both ice and chilled water storage systems. This means more cooling is taken from the storage tank, which also means that the storage capacity in the 50% demand limiting strategy is larger than the storage in the load levelling strategy. In the demand limiting operation, the power reduction that can be achieved depends upon the capacity to which the chiller is designed to be limited. Generally, the lower the capacity, the more power can be time shifted.

Full storage, as shown in Figure 6.2, is able to time shift the maximum electrical load compared with a conventional AC system and other operating strategies. During the discharging time, from 11:00 hours to 15:00 hours, the chillers in the full storage systems are entirely switched off, and the cooling load is directly met from the storage tank. The chiller and storage sizes are larger than those of other operating strategies. It has been reported that full storage can reduce the peak electricity demand by 80 to 90% (Hasnain, 1998).

6.3.3 Coefficient of performance

The results of the COP for the chillers selected in this report have been carefully analysed. Although the COP of a chiller increases with the decrease in the dry bulb temperature, it was also found that the COP depends upon the number of compressors in operation (for example, its percentage capacity); larger chillers have more compressors, and as the load on the chillers increases, the number of compressors in operation also increases.

For example, a chiller with three compressors has a high COP when the capacity of the chiller is either 33%, 60% or 100%, and a chiller with two compressors has a high COP when the capacity of the chiller is either 50% or 100% (see Figure 6.3). Moreover, it has been found that all the chillers selected in this report have only two compressors, and in some of these chillers, the COP at about 50% capacity is higher than is the COP of the chiller operating at full capacity. This indicates that during the part load operation, for example, at 50% capacity, only one compressor is on, and one is off, implying that the one compressor is operating at full load; hence, maximum efficiency is achieved as shown in Figure 3.3.

The COP of the chiller considers the total power input (for example, both compressor and condenser fan motors) to the chiller, not only the compressor power. Therefore, another factor that plays an important role in determining the COP is the number of condenser fan motors that come into operation with each compressor. This effect remains to be studied in more detail.

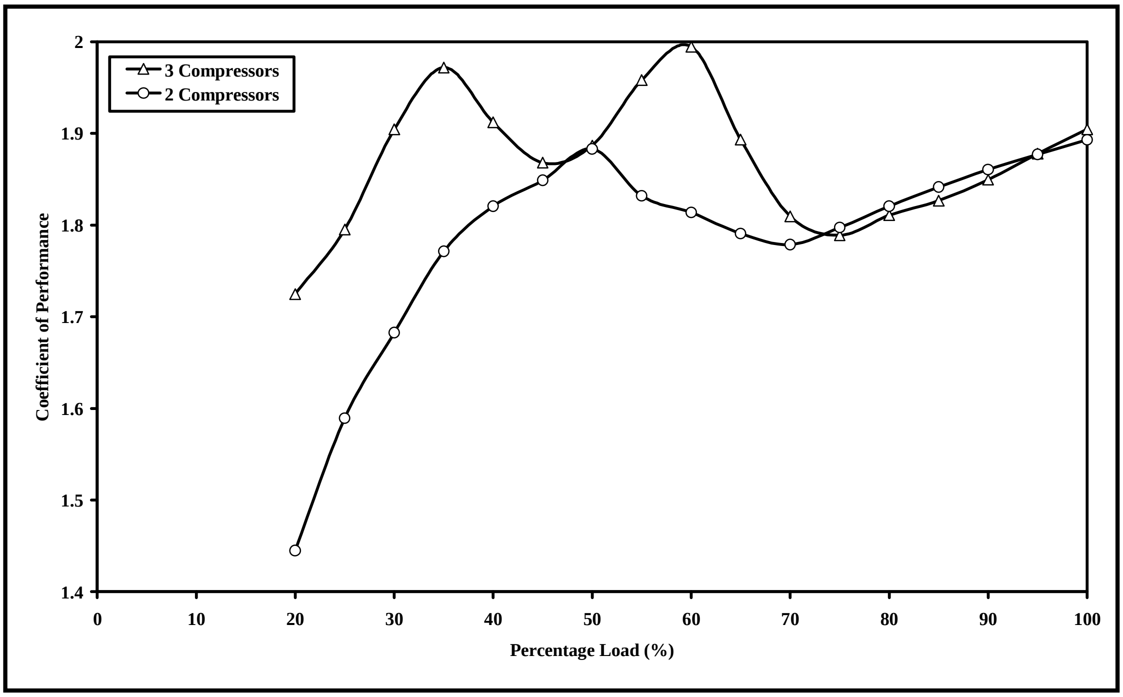

Figure 6.3 Plot of percentage chiller capacity vs. coefficient of performance at design condition.

The hourly profiles of the COP of the chillers are shown in Figure 6.4. The figure shows that the chiller operating in the conventional AC system has a higher COP during the night compared to during the day. It has been found that the average COP during the night time hours (for example, 18:00 hours to about 4:00 hours) is about 2.55 when the chiller operates at an average capacity of 37% full load (for example, one compressor is operating). However, the average COP during the day time (for example, from 8:00 hours to 15:00 hours) was found to be about 2.03 when the chiller operates at an average capacity of 76% full load (for example, 2 compressors are operating). The high value of the COP during the night time is due to the much lower dry bulb temperature compared to the day time dry bulb temperature leading to a lower condensing temperature and, in turn, a lower compressor discharge pressure.

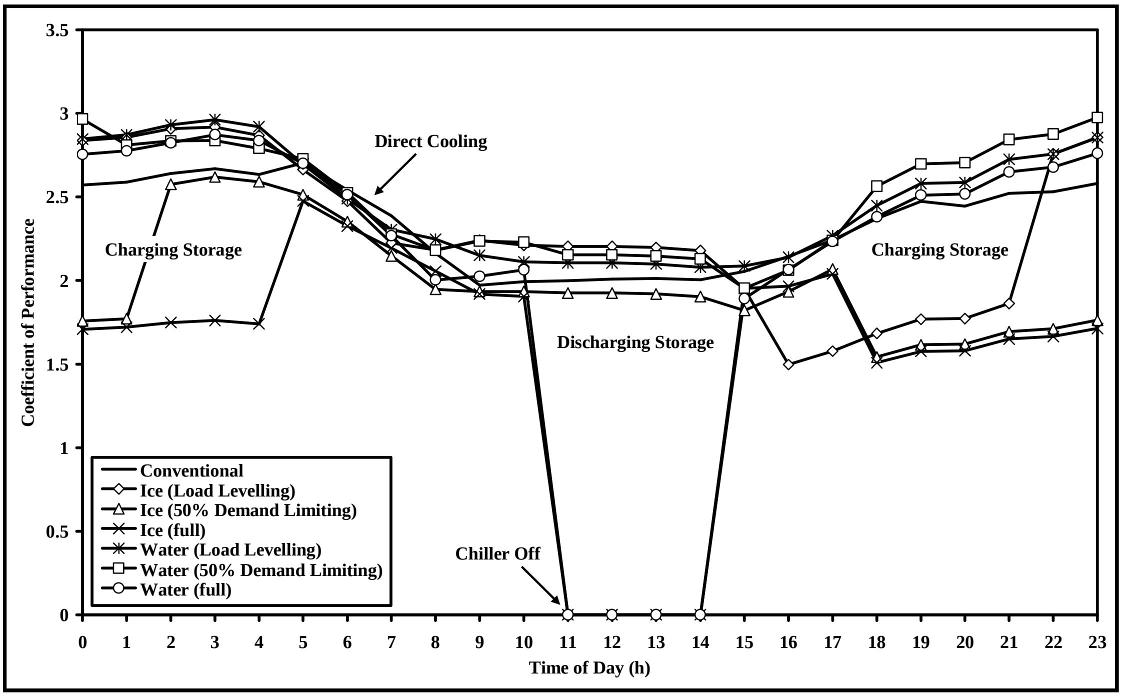

Figure 6.4 Hourly coefficients of performances profiles of the chillers at design condition.

Furthermore, for chillers operating with ice storage systems, it can be seen from Figure 6.4 that their COP during the charging time decreases significantly due to the much lower production of the chilled water temperature, which was -5.0° C compared to chillers in conventional and chilled water storage systems. It is estimated that the average reduction in the COP of the chillers during the charging time compared with normal operation at different dry bulb temperatures is about 24% for load levelling, 21% for 50% demand limiting, and 23% for full storage. Beggs (1992) in his design study assumed that the COPs of the chillers during charging time are about 23% lower compared to normal operation.

Figure 6.4 also shows that the chillers operating with chilled water storage systems have in fact the highest COP compared to the conventional and ice storage systems. In the load levelling operating strategy, the chiller operates at full load for most of the time and with an average COP of about 2.39 during full load operation. The COP of this chiller decreases during the day because of the high ambient temperature.

The chiller with 50% demand limiting strategy has two compressors; when the capacity of the chiller is limited to 50%, one compressor operates at full load and hence, at maximum efficiency. It was found that during the discharging time, the COP of the chiller operating at 50% is about 2.13. The chiller in a full storage system does not operate during the discharging hours, therefore, the COP is equal to zero as shown in the figure.

6.4 Breakdown of peak power and energy consumption

In this section, a breakdown of the components including the chillers, pumps and AHUs of the conventional, ice and chilled water storage AC systems for the maximum power demand and energy consumption is examined for the design day. For the maximum power demand, the breakdown of the components is estimated in terms of their proportion to the maximum power consumption, which occurs during the maximum cooling load (for example, at 14:00 hours), and for the energy consumption, the breakdown is estimated during the design day operation of the AC systems.

6.4.1 Power consumption

Table 6.1 shows the calculated numerical values of the power consumptions at peak cooling load of each component at each AC system design. The peak cooling load occurs at about 14:00 hours. Further discussions and comparisons of the results shown in the table are provided in this section.

Air Conditioning System	Chiller	Air Handling Unit	Pump	
			Primary	Secondary
Conventional	185	23	3	5
Ice (Partial Load Levelling)	141	23	6	4
Ice (Partial 50% Demand Limiting)	94	23	8	4
Ice (Full)	0	23	7	4
Water (Partial Load Levelling)	109	23	1	4
Water (Partial 50% Demand Limiting)	74	23	2	3
Water (Full)	0	23	0	4

Table 6.1 Calculated power consumption in kW_e of different components in the AC systems at maximum cooling load

The proportion of the power consumption of different components at the maximum cooling load in the various AC systems is shown in Figure 6.5. Compared to other components in the AC systems, the chiller in the conventional system has the largest power consumption. As has been mentioned before, in the conventional system, the maximum power consumption occurs at 14:00 hours, when the system load is highest. Furthermore, it can be seen from the figure that the chiller proportion to the maximum power in the conventional AC system is the highest compared to the ice and chilled water storage systems and represents about 85.5%.

It is also important to note that the proportion of the power consumption by the chiller in the conventional AC system is higher by 7.5% than the estimated power given by Maheshwari et al. (2003) as shown in Table 2.1 for the air-cooled system (refer to Chapter 2). This is because the COP of the chiller studied in this report has a COP of 2.04, which is lower than the one given in Table 2.1 by about 7.2%.

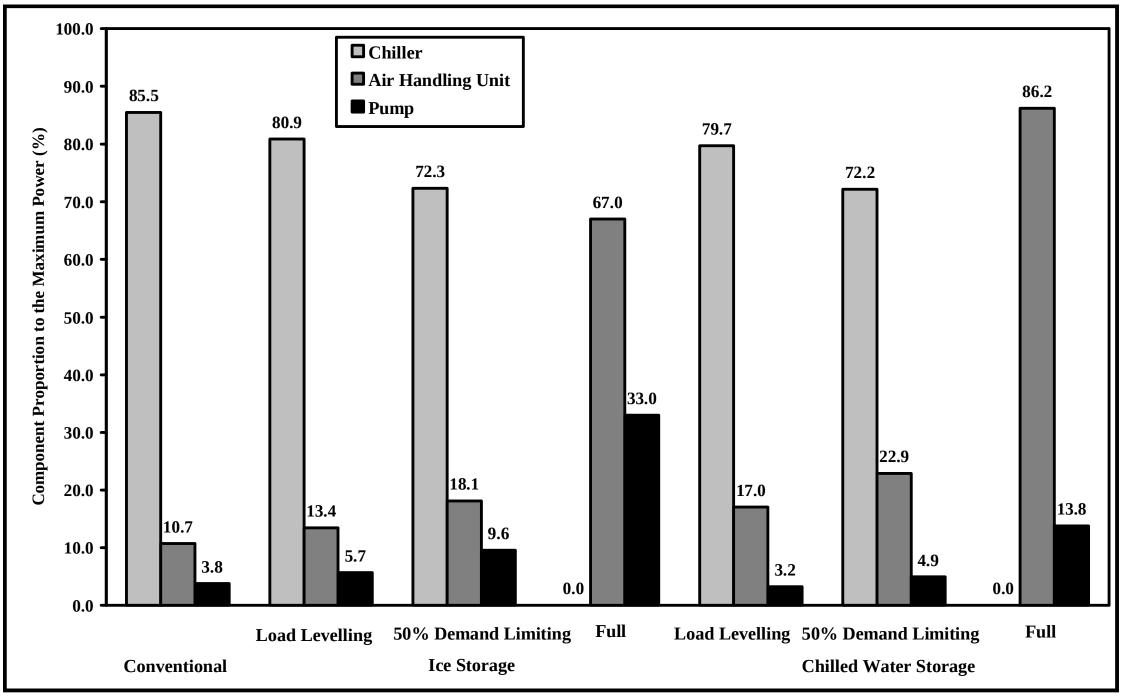

Figure 6.5 Proportion of various components in the AC system to the maximum power consumption.

It can be also observed that the chillers operating with full strategy in both ice and chilled water storage systems have zero proportion, hence they have the lowest power consumption compared to other operating strategies; this is because the chiller in this strategy is entirely switched off. Furthermore, the load levelling operating strategies have the second largest proportion of 80.9% and the proportion of the demand limiting strategies lies between load levelling and full storage strategies.

The proportion of the power consumption by the AHUs is the second highest compared to other components, and represents about 10.7% of the total power consumption in the conventional system. This, in fact, is much lower than the 18.5% given in Table 2.1, due to the increase in the proportion of energy used by the chiller resulting from the lower chiller efficiency in this AC system efficiency and due the selected AHUs having quite high motor and fan efficiencies. Furthermore, it is clear that the proportion of the power consumption by the AHUs and chilled water pumps to the maximum power consumption in ice and chilled water storage systems operating with full strategy are highest compared to conventional and other operating strategies, due to the chillers being switched off during the day.

Figure 6.5 also illustrates that the pumps have a lower proportion of total power consumption compared to other components in all AC systems except in the full storage strategy. In the conventional systems, the pumps' proportion is about 3.8% and is slightly lower than the estimated proportion of 3.4% as given in Table 2.1. The proportion of power consumption by the pumps increases when the AC system is operating with full storage strategy, and the pumps have the lowest proportion when the system is operating with only 3.2% of the load levelling strategy in the chilled water storage system.

6.4.2 Energy consumption

The summary of the energy consumption for each component at each AC system at the design day is given in Table 6.2. The proportion of the energy consumption given in the table is further illustrated in Figure 6.6.

Air Conditioning System	Chiller	Air Handling Unit	Pump	
			Primary	Secondary
Conventional	2297	559	66	98
Ice (Partial Load Levelling)	2431	563	133	78
Ice (Partial 50% Demand Limiting)	2730	563	194	78
Ice (Full)	2939	563	174	78
Water (Partial Load Levelling)	2156	561	21	64
Water (Partial 50% Demand Limiting)	2192	561	49	54
Water (Full)	2194	561	32	67

Table 6.2 Calculated energy consumption in kW_eh for different components in the AC systems at design day

It can be seen from the figure that in all AC systems, the proportions of energy consumption by the chillers are all above 70%, and range between a minimum of 75.8% for the case of the ice storage with a load levelling operating strategy to a maximum of 78.3% for the ice with full storage operating strategy.

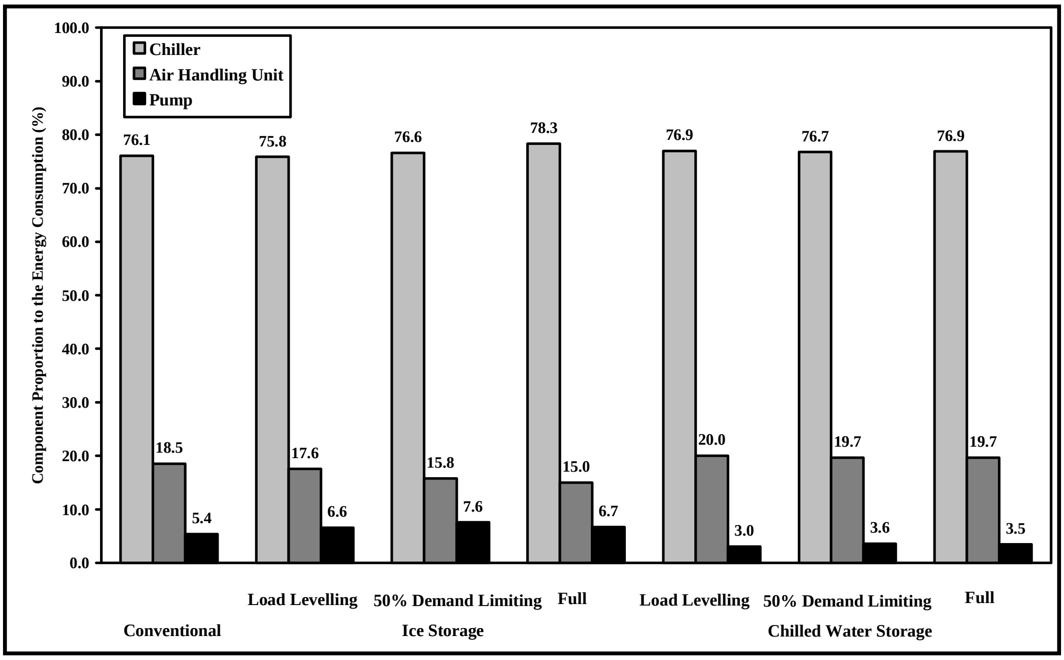

Figure 6.6 Proportion of energy consumption by various components in the AC system.

Since the operating cost is directly proportional to the electrical energy consumption and the cost of the electrical energy does not vary during the day, the chillers have the highest operating costs on peak summer days compared with other components in the AC systems. If the cost varies during the day, then the energy consumption must be calculated separately for the charging and discharging times and then the operating cost can be properly estimated (John, 1994).

Similarly, the proportion of total energy consumption by the AHUs in the AC systems does not vary significantly from one system to another except for 50% demand limiting and full ice storage systems, where the proportion slightly decreases to 15.8% and 15% respectively. This is due to the pumps in these two operating strategies having slightly larger energy consumption. Furthermore, compared to other components in the AC systems, Figure 6.6 shows that the pumps have the smallest proportion, which means they have the lowest energy consumption.

It should be noted that the above discussions of the proportion of the components to the power consumption and energy consumption as shown in Figure 6.6 are only applicable for the developed system load as given in Figure 5.7 in Chapter 5, and for other cooling load profiles that are similar to the system load profile, which happens during the day time in the summer months of June, July and August.

However, for other months in the year, for instance, January or December, the chillers and chilled water pumps are not operating; therefore, their proportion of the power and energy consumptions is zero. Similarly, in mild months such as April and November, the chillers and pumps consume less energy compared to the AHUs (assuming the AHUs are operating with constant speed motors); therefore, their proportions are also lower.

6.5 Comparison between alternative storage strategies

It is well known that ice and chilled water storage systems can reduce the power demand during the day time when the power demand of a conventional AC system is high. However, the amount of the reduction in the power that can be achieved when using storage systems differs from one operating strategy to another. Furthermore, another important factor that must be considered when the AC system is operating with ice and chilled water storage systems is the amount of increase or decrease in the energy consumption for each component as well as the overall energy consumption, because the operating cost increases in direct proportion to the energy consumption.

In the following subsections, one of the main objectives of this study is discussed, namely, the change in power and energy consumption for each component and the overall energy consumption between the conventional and the ice and chilled water storage systems.

6.5.1 Power demand

The change in the power consumption at the maximum cooling load for different components in the ice and chilled water storage systems for various operating strategies with conventional systems is shown in Figure 6.7. With ice storage systems, the overall reduction in the power is achieved with the full storage strategy where power decreases by 83.8% compared with conventional systems. Furthermore, load levelling operating strategies have the lowest reduction at 19.4%, and the reduction in the demand limiting strategy is 40.2%, somewhere between those of full and load levelling strategies. The high reduction in the overall power demand for full storage strategies is obtained due to the large power reduction of 100% established by the chiller.

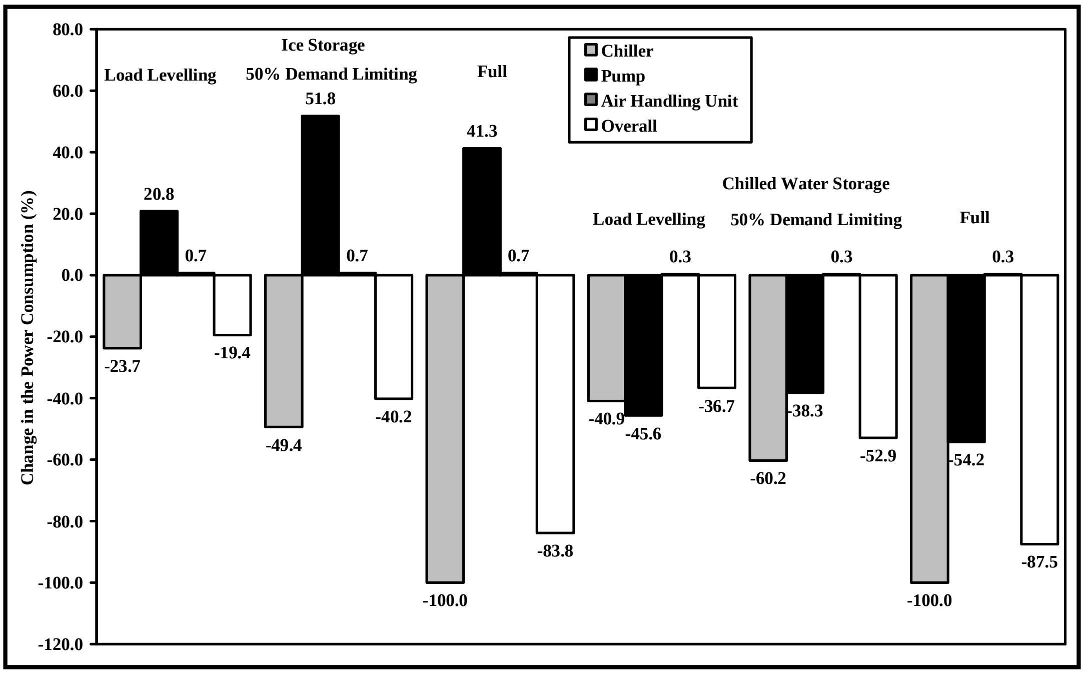

Figure 6.7 Change in the power consumption of ice and chilled water systems with conventional system.

Figure 6.7 also shows that the power consumption by the pumps in the ice storage systems increases between 20.8% to 51.8% depending on the operating strategy, compared with conventional systems. The increase of the pumping power is due to a large pressure drop in the ice tank, which was found to be about 11.5 m and an additional pressure drop associated with the circulation of a glycol solution in the pipes compared with the circulation of water.

Moreover, it can be observed from the figure that the power reduction in the AHUs is very small in both ice and chilled water systems compared with conventional systems; therefore, they make a negligible contribution to the reduction of the power consumption in the AC system. However, in ice storage systems, the air distribution system can be designed with colder air. With low air distribution, the connected power and energy consumption can be attained (Pearson, 1989; Meckler, 1989). There are some guidelines, and specific design considerations must be considered when the ducting system is designed with cold air distribution; these are reported in (Dorgan, 1989; (Harmon, 1989).

In chilled water storage systems, the full storage once more achieves the largest reduction in the overall power demand. It reduces the power demand by about 87.5% compared to conventional systems. For other operating strategies, load levelling and demand limiting reduce the overall power demand by 36.7% and 52.9% respectively. Furthermore, from Figure 6.7 it can be seen that the reduction in the power demand for chilled water storage systems is not only achieved by the chillers but also by the pumps. Compared to the conventional system, the reduction in the pumps' power ranges between 38.3% and 54.2% as a result of the lower pressure drop and the higher design temperature differential across the AHUs in the secondary piping systems for the chilled water storage systems.

6.5.2 Energy consumption

Although in the ice storage systems the overall power consumption is reduced compared with conventional systems, the overall energy consumption increases by 6.1% to 24.3% depending on the operating strategy as shown in Figure 6.8. The increase in the overall energy consumption is due mainly to the inefficient operation of the chillers during the charging time. The chillers in the charging time produce chilled water at a temperature of -5.0° C; consequently, the COP of the chillers significantly decreases to an average of 23% thus increasing the energy consumption of the chillers by 5.8% to 28% compared with conventional systems.

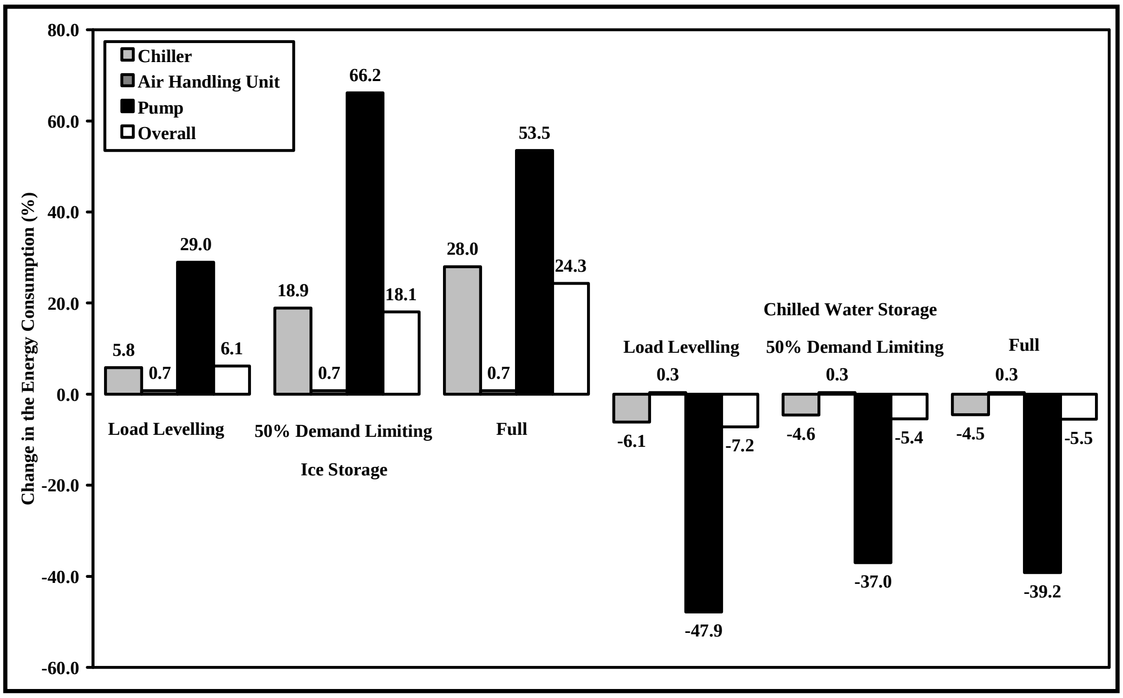

Figure 6.8 Change in the energy consumption of ice and chilled water systems compared with conventional systems.

The chilled water pumps in the ice storage systems are another factor that increases the overall energy consumption. Because the pumping heads in an ice storage system are much higher than they are in the conventional system, the pump sizes are therefore bigger and so more energy is consumed. The increase in the energy consumption of the pumps ranges from a minimum of 29.0% for the load levelling strategy to a maximum of 66.2% for the demand limiting strategy compared to that of a conventional system, and this is extremely high. Figure 6.8 also shows that the energy consumption by the AHUs in both ice and chilled water storage systems are small since the sizes of the AHUs are approximately the same for all AC systems.

Chilled water storage systems are unlike ice storage systems; they can save energy as illustrated in Figure 6.8, therefore they are more attractive for the conditions in

Kuwait. The achieved reduction in the pumping energy is between 37% and 47.9% and energy consumption by the chillers between 4.5% and 6.1% depending on the operating strategy, which results in a reduction of overall energy of 5.4% to 7.2% compared with a conventional system. This can be considered one of the main and most significant advantages of using chilled water storage system technology compared with ice storage system technology.

6.5.3 Further analysis

Further analysis has been conducted based on the selection of the charging hours in an attempt to reduce the energy consumption of the chillers for different operating strategies in ice and chilled water storage AC systems. For example, the results obtained in Section 6.4.2 for the operation of the chillers' storage systems for demand limiting partial and full storage operations were based on the selected charging hour starting at 18:00 hours.

The energy consumption of the chiller could be reduced by shifting the charging time by three or four hours later to take advantage of the lower dry bulb temperature during the night as well as the lower system load. Similarly, for partial load levelling, the charging time was shifted from 15:00 hours to 22:00 hours in the ice storage system and from 15:00 hours to 19:00 hours in the chilled water storage system. The following improvements in energy consumption in the chillers were established:

a. In ice storage with load levelling strategy, the energy consumption is reduced by 2%.

b. In ice storage with demand limiting strategy, the energy consumption is reduced by 2%.

c. In ice storage with full strategy, the charging hours remained unchanged; therefore, there is no improvement in energy consumption.

d. In chilled water storage with load levelling strategy, there is no improvement in the energy consumption.

e. In chilled water storage with demand limiting strategy, the energy consumption is reduced by 1%.

f. In chilled water storage with full strategy storage the energy consumption is reduced by 2%.

By calculating the power consumption of the chillers based on the implementation of the above time shifting, and adding the power consumptions of the AHUs and pumps, the hourly overall power consumption profile for the design day is developed for each AC system and plotted as shown in Figure 6.9.

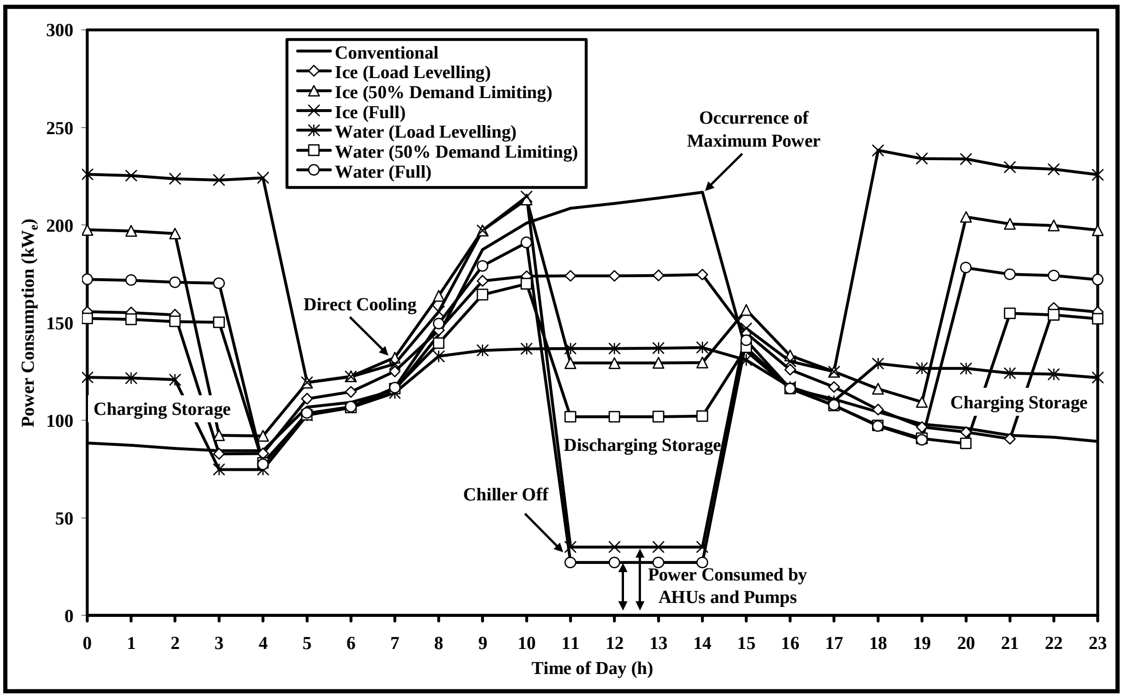

Figure 6.9 Hourly overall power consumption profiles for the design day.

It is clear from Figure 6.9 that the maximum power demand occurs at 14:00 hours for the conventional system. For other operating strategies, in the ice and chilled water storage systems the power consumption is reduced during this time.

6.6 Annual energy analysis

In this section, another objective of this thesis is examined, namely, the proportion of energy consumption used by different components to the overall annual energy consumption and the change in the annual energy consumption of the ice and chilled water storage systems compared with a conventional AC system. The annual energy consumption in storage systems is calculated as given in Table 6.3 using the typical meteorological year that was developed for Kuwait conditions (Shaban, 2000) and based on the improved energy consumption resulting from shifting the charging time as discussed in Section 6.5.3 above.

Air Conditioning System	Chiller	Air Handling Unit	Pump	
			Primary	Secondary
Conventional	307	204	12	15
Ice (Partial Load Levelling)	290	206	29	12
Ice (Partial 50% Demand Limiting)	344	206	35	12
Ice (Full)	392	206	32	12
Water (Partial Load Levelling)	286	205	4	10
Water (Partial 50% Demand Limiting)	279	205	9	8
Water (Full)	291	205	7	10

Table 6.3 Annual energy consumption in MW_eh of each component in the AC systems

6.6.1 Component proportion

Figure 6.10 illustrates the proportion of energy consumption by the AC components including chiller, AHU and pump to the overall annual energy consumption for various AC systems. For all AC systems, the chillers use the largest proportion of the annual energy consumption, which ranges between 54.1% to 61.0% depending on the design of the system; an ice storage system operating with full strategy uses the largest proportion of 61.0%. This high proportion of the chiller to the annual energy consumption is mainly due to the high energy consumption by these chillers during their operation in the peak summer months of June, July, August, and September when the ambient day time temperature is usually above 40° C and the night time temperature above 30° C.

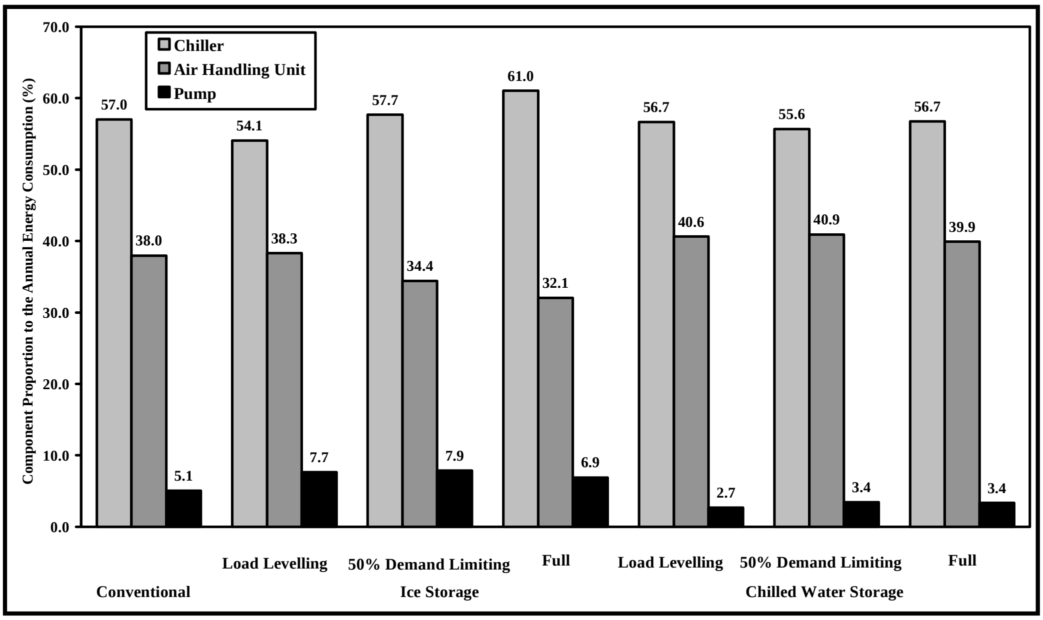

Figure 6.10 Proportion of the energy consumption by components in the AC systems to the annual energy consumption.

Figure 6.10 also shows that the AHUs also use a large proportion of the annual energy consumption; their proportion ranges between 32.1% and 40.9% depending on the design of the AC system. The high proportion of energy used by the AHUs is because of their continuous operation throughout the year irrespective of the building load. Moreover, in the winter, for example, in the months of January, February and December, AHUs are the only components operating in the AC systems, which mean the proportion of the energy consumption used by them at these months is 100%.

The pumps have the lowest proportion compared with other components in the AC systems. In the ice storage systems, the pumps use a higher proportion compared with conventional and chilled water storage systems. The proportion used by the pumps in the ice storage systems operating with full strategy is 7.9%, which is the largest. Furthermore, the proportion used by the pumps in the chilled water storage systems is lower than that of both the conventional and ice storage systems, as the proportion reaches a maximum of 3.4% when the chilled water storage system is designed with either a 50% demand limiting or full storage operating strategy.

By referring to Figure 6.10, the operating cost of each component in the AC system can be estimated. For example, the chiller in the conventional AC system represents about 75% of the total operating cost per year, while AHUs represent 38% and pumps about 5.1%.

6.6.2 Energy comparison

One of the most important factors that affect the operating cost, and hence the life cycle cost, is the annual energy consumption. Therefore, it is very important to study the comparison of the annual energy consumption of ice or chilled water storage systems with that of conventional systems. Generally, if the ice or chilled water storage systems save energy, this means they can also save on operating costs, which makes the storage technology more attractive compared with that of conventional systems.

Figure 6.11 shows the comparison in the annual energy consumption for the AC components and the overall energy consumption of the ice and chilled water storage systems compared with that of a conventional system. It can be seen from the figure that in all operation strategies, the ice storage systems consume more energy than do the conventional systems. The figure shows that the overall energy consumption in the full storage strategy system is 19.3% more than that of a conventional system, and this is due mainly to the higher energy consumption of the chiller (27.7%) and pumps (63.4%).

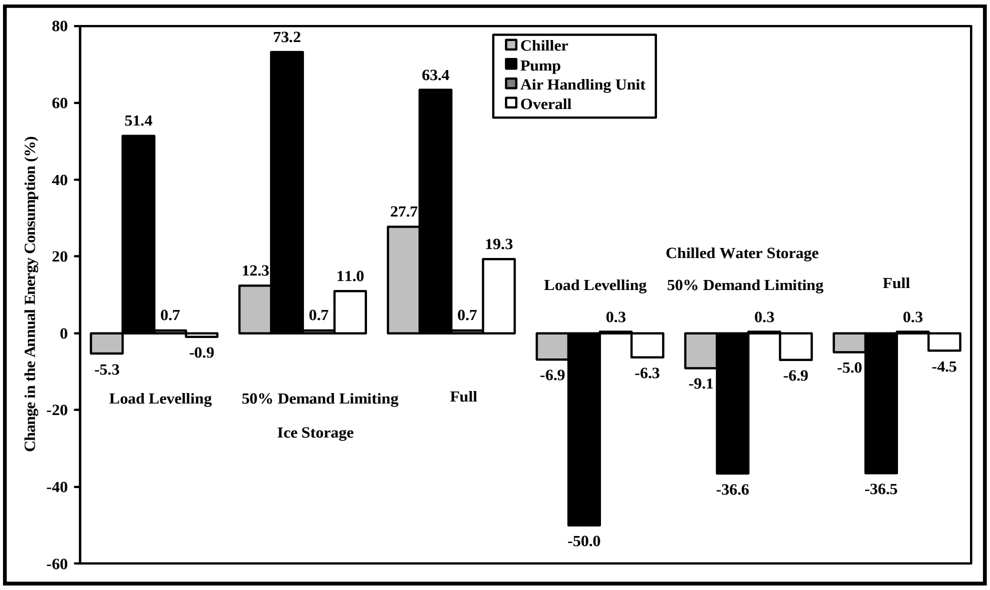

Figure 6.11 Change in the annual energy consumption of the ice and chilled water storage systems compared with that of the conventional system.

Likewise, the overall annual energy consumption of the ice storage system operating with a demand limiting strategy also consumes more energy, and is 11.0% higher than that of the conventional system. In this operating strategy, the chiller consumes 12.3% more energy and the energy consumption by the pumps is 73.2% higher than in a conventional system. However, in a load levelling strategy, a reduction of 0.9% is achieved as shown in the figure.

For the chilled water storage system, the case is different; the reduction in the overall energy consumption is established for all operating strategies as shown in Figure 6.11. It is clear that the highest reduction is realised with a demand limiting strategy of 6.9%, which is due to the reduction in the energy consumption of the chiller (9.1%) and in that of the chilled water pumps (36.6%).

The reduction in the overall annual energy consumption for the load levelling strategy is 6.3% and for the full storage operating strategy is 4.5%. In addition, the energy consumption of the pumps in the load levelling strategy is reduced by 50% compared with that of the pumps in the conventional system. It is also clear from Figure 6.11 that the AHUs have the least effect on the annual energy consumption; they increase the consumption to about 0.7% in ice storage systems and 0.3% in chilled water storage systems.

By further analysing the data obtained for the energy consumptions in mild months such as April, May, October, and November, it was found that the chillers and pumps in ice storage systems operating with demand limiting and full strategies consumed more energy compared with those of a conventional AC system, and this was found to contribute further to the increase in the overall annual energy consumption by these systems. In fact, the main reason for the increase in energy consumption by the chillers in the ice storage system, particularly those operating with demand limiting and full strategies, is due to the lower COP of these chillers compared with that of the chiller in the conventional system. See Appendix B for the chillers' specifications. Another reason is that the heat gains by the auxiliary systems in the ice storage systems are slightly higher than in the conventional system.

In chilled water storage systems, the chillers were found to have a slightly higher COP except for the full storage strategy where the COP of the chiller is slightly lower.

Because the heat gains by the auxiliary systems are lower than are those of the conventional system, the energy consumption of the chilled water systems was found to be lower in the mild months. Furthermore, in the mild months, the pumps consumed less energy compared with the conventional system.

6.7 Conclusions

In this chapter, the power requirements for each component in the conventional, ice and chilled water storage AC systems was calculated for the design day and for the whole year. The calculated power and energy consumptions were studied and analysed. The proportion of power and energy consumptions for each component for the design day conditions of Kuwait were estimated. Furthermore, the energy use by the components was also estimated in proportion to the annual energy consumption using the typical meteorological year.

For the design day conditions, it has been shown that chillers, in all AC systems except full storage strategy systems, use the largest proportion of power consumption, from between 72.2% and 85.5% at the maximum system load. They also had the largest energy consumption from between 75.8% and 78.3% compared with other components in AC systems. The second largest proportion in the power consumption was used by the AHUs at between 10.7% and 86.2.9%, and in the energy consumption from between 15.0% and 20% depending on the type of the system and operating strategy. Moreover, the pumps had the lowest proportions.

Furthermore, it has been found that the use of ice storage systems reduced the overall power consumption by between 19.4% and 83.8%, and increased the energy consumption by between 6.1% and 24.3% as compared with conventional systems, and in the chilled water storage the power consumption was reduced from between 36.7% and 87.5% and the energy consumption decreased by between 5.4% and 7.2% depending on the selected operating strategy. The large reduction in power consumption was achieved by the operation of the chillers. It was also found that the chillers and chilled water pumps in the ice storage systems increased the energy consumption; however, in the chilled water system, the power and energy consumption of the chillers and pumps were reduced.

It has also been found the overall annual energy consumption of the ice storage systems operating with 50% demand limiting was increased by 11.0%, and by 19.3%

when was operating with full strategy. However, with a load levelling strategy the consumption decreased by 0.9%. The increase in the overall annual energy consumption was due mainly to the increase in the energy consumption of the chillers and pumps.

With chilled water storage systems, a significant reduction in the overall annual energy consumption was established. The overall energy was reduced by between 4.5% and 6.9% compared with conventional systems depending on the operating strategy; the chillers and pumps were both contributed to this reduction.

With the detailed discussion provided in this chapter, we may conclude that AC systems operating with chilled water storage have reduced both the power and energy consumption; therefore, they can be considered the optimal storage technology choice for the Kuwait climate. Furthermore, if a high reduction in power consumption is required, the building owner can design the system with full strategy storage. A full storage system also has the advantage of being applicable to the future expansion of the cooling system, as it can be operated as a load levelling or demand limiting partial storage system without additional cost. In addition, if the lower reduction in the power consumption is required, the building owner can design the system with either load levelling or 50% demand limiting strategies. However, for the proper selection of storage technology and operating of the cool storage system, a life cycle cost provides a more detailed analysis for the optimal choice and this will be the subject of next chapter.

Chapter 7

Economic Analysis

7.1 Introduction

Life cycle cost is an economic evaluation technique that determines the total cost of owning and operating a system over a period of time and it is essential for controlling the initial and the future cost of the ownership (Sieglinde, 1996). The main purpose of performing a life cycle cost analysis is to make good decisions if different designs or an improvement of a system is implemented.

One of the objectives of this thesis is to investigate the economic impact of using ice and chilled water storage on the AC systems. In this chapter, the economics of conventional AC and cool thermal storage with ice and chilled water operating with load levelling, 50% demand limiting and full storage strategies are assessed. The assessment is based on the actual and subsidised costs of electricity consumption and connected electrical power.

This chapter discusses the procedures for computing the total life cycle cost for each AC system alternative to evaluate the total cost of each systems over its complete life time and hence to compare the results in order to select the most economic alternative. Each term in the life cycle cost analysis is determined including system capital, operating, and maintenance costs. The capital costs for the various components on the AC systems are obtained from local suppliers and contractors agencies. The breakdown of the capital cost of each AC system is tabulated in Appendix G. The results for the capital costs for the AC systems are also analysed and discussed in this chapter.

Furthermore, the estimation of the operating and maintenance costs, which plays an important role in evaluating the life cycle cost of the AC systems, is given. This chapter will also present the best selection of the AC systems that require the minimum life cycle cost.

7.2 Capital cost

The fist step in the evaluation of the life cycle cost is to define all the initial investment costs for the conventional, ice storage and chilled water storage AC systems. The costs for each system were obtained based on the sum of the individual cost of the following subsystems,

a. Cooling production.

b. Storage.

c. Air distribution.

d. Water distribution.

The costs of the above systems are discussed in more detail in the following subsections.

7.2.1 Water distribution

Pumps, pipes, and valves including pipe insulations are the main components in the chilled water distribution system. Their costs generally increase with the size of the AC system; larger systems require larger pipes, valves and pumps, hence, the high water distribution cost.

Moreover, when the AC system is designed with ice or chilled water storage systems, there will be additional piping and valves required and the size of the chilled water pumps may be higher, which means the cost of the water distribution system will increase compared with that of a conventional AC system. Appendix G lists the costs of the water distribution systems that were obtained from different suppliers. Further discussion and factors affected the water distribution system are given below.

7.2.1.1 Pumps

The cost of the pumps depends on the sizes and types, and on pump and motor efficiencies. The size of the pumps increases with the increase in the pumping head and design volumetric flow rate. The pumping heads in the primary and secondary chilled water pumps in the AC systems were estimated after the pressure drop in the chillers, AHUs, ice and chilled water storage and pipes was calculated (see Table D.3).

The design volumetric flow rates of the chilled water in the primary and secondary circuits were determined as discussed in Appendix D and summarized in Table D.1. The final sizes of the pumps were then obtained, and then their costs were obtained from the pumps suppliers, which were also based on their motor and pump efficiencies (see Appendix B).

7.2.1.2 __Pipes__

The cost of the piping systems was found to depend strongly on the following factors (Buys, 2005):

a. Site and piping layout.

b. Design of the piping system.

c. Pipe material and sizes.

d. Pipe insulations.

e. Valves and other accessories.

The site and piping layout are closely related. The distance between the plant room where the cooling is produced and the AHUs has a significant influence on the length of the pipes and therefore on their initial and installation costs. The piping layouts in the plant room and on the roof of the building for conventional, ice thermal storage, and chilled water storage are discussed in Appendix D.

The piping design has a significant effect on the overall cost of the water distribution system. The piping systems were arranged with primary-secondary piping designs (James, 1996; ASHRAE, 2000c) (see Appendix D). This design is considered today to be the most popular chilled water system because it separates the chiller (that is, the chilled water production zone) from the distribution piping system (that is, the chilled water transportation zone) thereby reducing the differential pressure drop across the control valves of the cooling coil.

In most installations, seamless black steel heavy duty schedule 40 pipes are used, therefore the cost in US$ per metre of the pipes was determined based on the calculated length and nominal diameters. The nominal sizes of the pipes in the primary circuits depend on the design chilled water flow rate of the chillers, and in the secondary circuits, they depend on the design flow rate of the AHUs. The sizes of the pipes were calculated based on a recommended flow rate and maximum pressure drop per pipe length (refer to Section D.3 of Appendix D).

Moreover, the pipe lengths in the secondary circuits were the same for all systems. However, for the primary circuits, the lengths of the pipes in the ice and chilled water storage systems were greater compared with those in the conventional system because of the extra pipes associated with the ice tanks and with the diffusers in the chilled water tanks. Additional costs were added to the cost of the pipes to account for the piping supports, hangers, and sleeves. These costs were estimated based on the costing analysis for three existing projects provided by a contractor.

Another important expense that must be considered in the piping system is the pipe thermal insulation. Thermal insulation has a significant influence on the piping costs. The cost varies depending on the insulation material (that is, insulation density) and thickness. The cost of the insulation was determined for pipes insulated with rigid fibreglass of 96 kgm^{-3} density for 25 mm and 50 mm insulation thicknesses. In addition, the insulation of the piping is commonly protected with cladded galvanized steel, aluminium or stainless steel; therefore, additional costs were added to account for the cladding.

7.2.2 Air distribution

The air distribution system consists mainly of fans, ducting, flexible, terminals and insulation. Here, it was also assumed that the AHUs are part of the air distribution systems, as the AHUs contain the supply fans and sometimes return fans that circulate the air in the ducts. In the following subsections, the costs of the AHUs and ducts for conventional ice and chilled water storage AC systems are discussed.

7.2.2.1 <u>Air handling units</u>

AHUs consist of fans, cooling and heating coils, electrical heaters, casings and filters (ASHRAE, 2000a; ASHRAE, 2000b; ASHRAE, 2000d). The overall cost of the AHUs depends on the individual cost of each item and its specification, which will be discussed in this section.

All the costs of the AHUs were obtained from the supplier based on the following given specifications,

a. Mixing box with outdoor air and return air dampers and louvers.
b. Panel filter low efficiency.
c. Bags filter medium efficiency.
d. Cooling coil.
e. Supply double inlet airfoil or backward inclined centrifugal fan.
f. Horizontal and draw through unit.
g. Double wall sheet metal casing, insulation (25 mm – 50 mm).
h. Fan efficiency minimum 60%.
i. Motor efficiency minimum 80%.

The AHUs based on the above specifications were selected using the FLEX-air Version 5.08SA software program (Carrier, 2005). In the selection program, the mixing box was first selected for the AHUs. It was selected with a return and fresh air dampers and louvers; the locations of the air inlets for both fresh and return airs were also specified.

The second section of the AHUs that was selected was the filter section. The main function of the filters is to achieve an acceptable indoor air quality for the building. Two types of filters were chosen, low efficiency panel and medium efficiency bag filters. Low efficiency panel filters are often used to remove dust particles sized between 3 µm and 10 µm and medium efficiency filters are used to remove dust particles of sizes 1 µm to 3 µm (ASHRAE, 2000a; Shan, 2001). Generally, the cost of these types of filters depends on their sizes, types, and efficiencies.

The third section that was added in the AHUs was the water cooling coil. The construction and characteristics of the cooling coils are discussed in detail in the ASHRAE handbook (ASHRAE, 2000b). Basically, the cost of the coils depends on their configuration and construction, for example,

a. Fin spacing.
b. Number of rows.
c. Height and width.
d. Tube diameter.

e. Material construction.

The specification of the water tubes were copper tubes of 16 mm diameter with a thickness of 0.5 mm, and that for the fins was copper with a stainless steel casing. The final configuration of the cooling coils (for example, number of rows, number of fins per mm face area and so on) were automatically calculated by the program based on the input data such as coil dry and wet bulb temperatures, total volumetric flow rate of air, inlet chilled water temperature, and temperature differential across the cooling coils (see Appendix D).

The final section of the AHUs was made for the supply fans and their motors. Draw through units were selected because the discharge air can be easily connected to the supply duct and in fact they are most widely used (Shan, 2001). Other data specified for the fans was the fan discharge air locations, which were assumed to be at the rear and top horizontal front respectively.

In order to complete the proper selection for the fans, the data for the design volumetric flow rate of the supply air and upstream external static pressure was inputted. The upstream external static pressure (for example, pressure drop in the ducts) for all AC systems was assumed to be the same since the configurations and flow rates in the ducting system were the same. However, the pressure drop across the cooling coils, filters, and mixing boxes was calculated by the software and differed because of the difference in the temperature differential across the cooling coils between the conventional and ice and chilled water storage systems. This resulted in a difference in the coil configuration and face velocity and hence the pressures drop across each item in the AHUs.

Furthermore, a double inlet airfoil and backward-inclined centrifugal fan was selected for the AHUs since it is often used for large AHUs with a greater air flow rate and higher fan total pressure due to its higher efficiency and lower noise (ASHRAE, 2000d). A forward curved centrifugal fan was also available; however, they are often more suitable for small AHUs and where the volumetric air flow rate and fan total pressure are lower (ASHRAE, 2000d).

In addition, there are two kinds of casing for the AHUs and both are widely used: a double wall sheet metal and single sheet metal panel with inner insulation. The insulation material in a double wall sheet metal is sandwiched between two sheet metal panels of approximately 25 mm to 50 mm thickness with an overall heat transfer coefficient of 0.68 $Wm^{-2}°C^{-1}$ to 1.42 $Wm^{-2}°C^{-1}$ (Shan, 2001).

The cost of the casing depends on the air flow rate and the number of individual sections required for housing (for example, mixing plenums, filters, coils, heater and fans). Since the recommended velocity through the AHUs was selected to be in the range of 2 ms^{-1} to 2.5 ms^{-1} as recommended by ASHRAE (ASHRAE, 2000d), the sizes of cross sectional areas of the AHUs were determined by the volumetric flow rate of the air, which was already calculated and which is listed in Table B.4 of Appendix B. Furthermore, the length of the casing was determined by the type and number of the individual sections on which the AHUs were built.

It is very important to determine properly the dimensions of the AHUs as well as their number to be able to specify the cost of the transportation accurately. For the clinic, the transportation cost remained the same since the dimensions of the AHUs in the conventional, ice and chilled water storage AC systems were approximately similar. For the AHUs in each system, the transportation cost was estimated to be about US$ 2.1 thousand. The costs of the AHUs including their transportation to the site (excluding off loading) are listed in Tables G.1 to G.7 of Appendix G.

7.2.2.2 __Ducts__

The design of the ducting system for all AC systems was assumed to be the same; hence, the cost was also the same. The cost was obtained from a leading AC contractor and was estimated based on a review of the ducting layout provided by the AC design engineer of the building. The cost of the ducting system was estimated to be about US$ 32.4 thousand; the cost included ducting installation (for example, galvanized sheet duct, hangars, supporters, dampers, insulations, diffusers, grills and so on), labour, overheads, and profits.

7.2.3 Cooling production

The cost of the cooling production in the AC systems is always largest compared with other system costs. In the conventional air cooled AC system, the equipment in the cooling production system consists only of a package air cooled chiller; therefore, the cooling production cost depends on the size, number and installation of the chillers. In the ice and chilled water storage AC systems, ice and chilled water storage tanks are additional equipment. The cooling production costs were determined including the costs of the packaged chillers, ice tanks and chilled water storage tanks, as well as their installation costs, including contractor overheads and profit.

7.2.3.1 <u>Chillers</u>

Generally in Kuwait, air cooled chillers with reciprocating compressors are about 25% more expensive than are those with crew compressors, and their costs in US$ per Kilowatt cooling initially drops to between 176 kW_t and 422 kW_t and then stays constant (Al-Taqi, 2000). The cost of the chillers for the models selected by ECAT2 Version 4.12 (Carrier, 2004) chiller selection software was obtained from the supplier; their costs were determined based on the given specifications listed in Appendix B.

The transportation costs were also included, and they represented about 5% to 10% of the chiller cost depending on the chiller size; therefore, it was important to include the transportation costs in the capital cost estimation. Essentially, the transportation cost depends on the sizes (that is, dimensions) and number of the chillers that are transported to the site, and on the location of the country in which the chillers are manufactured and from which they are supplied. For all chillers listed in Appendix B, the transportation cost was estimated to be about US$ 5.3 thousand; this represents the cost of a container with a length of 6.10 m, width of 2.34 m and height of 2.29 m.

Furthermore, it was found that the chillers with a lower cooling capacity have a higher US$ cost per cooling capacity especially for very small chillers such as the selected chiller in the chilled water storage system with load levelling strategy. Moreover, based on the discussion with the supplier engineer, the cost slightly increases when the chiller is selected to produce a lower chilled water temperature such as in the case of ice storage systems because of the additional control requirement.

7.2.4 Electrics and control

Based on the costing analysis of three existing projects for conventional AC systems obtained from the contractor agency, the cost of the electrics and control including labour overheads and profit were found to be between 5% and 10% of the overall AC system cost, and this was also verified by (Buys, 2005). Therefore, it must be included in the analysis of the life cycle cost if an accurate estimation is required. In the case of ice and chilled water AC systems, the cost may be slightly higher than in the conventional system because of more additional components, control valves and sensors associated with these types of systems. In the following subsections, the costs of the electrics and controls will be further discussed.

7.2.4.1 __Electrics__

The cost of the electrical work mainly depends on the site layout and the equipment location in relation to the power supply points; therefore, it varies from one AC project to another. Basically, the electrical system of an AC system consists of the electrical control centre, the distribution board, and wiring.

The electrical control centre includes electrical switchgears for the isolating, protecting and switching of electrical equipments (for example, electrical heaters and motors in the AHU and compressor motors) in addition to the incomer section, which consists of a main isolator or fuse switch feeding power to all the switchgear in the control centre (Buys, 2005).

The cost of distribution boards and cables can be estimated if the number and sizes of the electrical control centres are known. The cost distribution boards can be determined based on a number of feed and amperage ratings, and the cost of the cables can be obtained based on the amperage ratings and cable length.

Alternatively, if such data discussed above are not available, the cost of the electrics (that is, electrical work) can be estimated in US$ per kW_e connected power (Maheshwari, 2003). Maheshwari (2003) estimated the cost of the electrical power to be 86.5 US$ per kW_e; his estimation was based on consultations with several contracting agencies in Kuwait. The connected power of different components in the AC systems was calculated from their specifications obtained from the suppliers and listed in Appendix B.

7.2.4.2 __Control__

The primary function of the control system in the AC is to modulate the capacity of the equipment to maintain predetermined parameters within the building or fluid entering or leaving the equipment to meet the cooling demand and climate changes at optimum energy consumption and safe operation. A control loop is the basic element of the control system, and consists of a sensor (for example, temperature or flow rate) that senses and measures the control variable, and a direct digital controller, which compares the sensed input signal with the set point and sends an output signal to actuate a controlled device such as dampers or a valve (ASHRAE, 2001b).

The cost of the control system and monitoring makes up 5% to 15% of the cost of an AC installation (Buys, 2005); it depends on many factors but mainly on the complexity of the control and type and number of the control devices, sensors and their locations and controllers.

The costs of the control systems for the AC systems were estimated by the contractor. The estimated costs were obtained by providing the contractor with the details of the temperature, humidity, flow rate and so on, the sensors and their locations on the piping system, the number of control devices including different types of valves and dampers, and the number and type of the input and output that actually control the systems. See Appendix I. Based on the above information, the contractor was able to determine the model and types of the controllers and their costs based on the number of input and output signals. The costs of the building automation system, installation, wiring, sensors, conduit, and so on were also included in the capital cost of the AC systems.

7.2.5 Storage

In ice and chilled water storage AC systems, ice and chilled water storage tanks are additional equipment; therefore, their costs must be added to the overall capital cost of the system. Generally, the cost of the storage tanks depends on the type of storage medium, tank construction and their number and sizes. Commonly, the cost of the ice tank in US$ per kW_t is higher compared with the cost of the chilled water tank; however, for the same application, the size of the chilled water tank (that is, tank volume) is much larger than that of the ice tank, because the chilled water storage relies on the sensible heat capacity of the water to store cooling whereas the ice storage system relies on latent energy.

7.2.5.1 Ice storage

Commonly, ice storage tanks are available as modular units from several domestic manufacturers (DUNHAM-BUSH, 1995; CALMAC, 2001b). The unit cost of ice storage varies with the type of storage technology and manufacturer, although most of the manufacturers are from the US. Moreover, their costs are strongly dependent on size especially in the smaller capacity range (EPRI, 1987).

The type of ice storage systems considered in this report is of internal melt ice-on-coil (see Section 3.5); their cost, including the cost of transportation, was obtained from the supplier in Kuwait (DUNHAM-BUSH, 1995). For the ice tank (that is, the smallest size) in the load levelling strategy, it was found that the cost in US$ per Kilowatt cooling was 65, which is very high compared with the cost of 31.4 for the 50% demand limiting and 27.8 for the full strategies.

7.2.5.2 Chilled water storage

The cost of the chilled water storage tanks varies depending on the material, shape, location, insulation and, more importantly, on the size of the tank. The cost of the storage tanks (excluding diffusers and pipes) in the chilled water storage systems (see Appendix D) was obtained from a cylindrical steel tank manufacturer; their cost was estimated based on the following data provided to the manufacturer.

a. The tanks have a cylindrical shape with dimensions given in Appendix D.
b. The tanks should withstand a maximum pressure of 500 kPa.
c. The tanks are above ground steel tanks and are constructed at the site.

Furthermore, the cost of the insulation and aluminium cladding was also included with the cost of the steel tanks and was approximately 12% of the total tank cost for the load levelling and 50% for the demand limiting strategies, and about 10.9% for the full storage strategy.

7.3 Capital cost analysis

It is important to examine the capital cost of the AC subsystem to identify the most expensive items in an AC installation. For the clinic, the capital costs of the conventional, ice and chilled water storage AC systems were analysed. The results are shown in Figure 7.1. The cooling production (that is, the chillers) as shown in the figure have the highest capital cost compared to other subsystems their costs range between 39.4% and 51.2% depending on the AC system design. It is also clear from Figure 7.1 that the second highest subsystem cost is that of the air distribution, which contributes between 23.4% and 27.8% of the overall AC capital cost. The water distribution and control and electrics subsystems have the lowest cost proportions.

Moreover, Figure 7.1 also illustrates that the cost proportion of the storage subsystem to the capital cost is not small, particularly with the chilled water storage AC systems where the storage cost reaches about 14.3% with the full storage strategy.

Furthermore, it can be seen that the cost proportion of the chillers in ice and chilled water storage systems is lower than is those of the conventional system, because of the smaller chiller sizes in these systems and the additional cost of the storage tank.

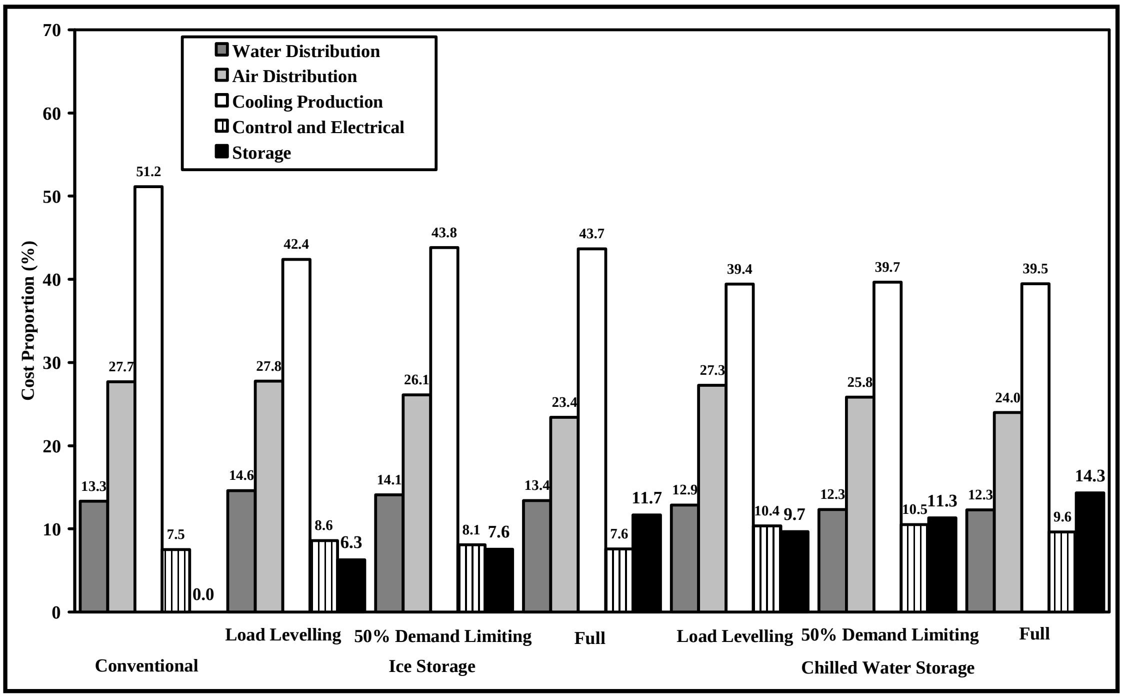

Figure 7.1 Proportion of various subsystems to the capital cost of the AC system.

It is also essential to examine the change in the capital cost of the individual subsystems of the ice and chilled water storage systems compared with the conventional system. This examination further illustrates the impact of using the cool thermal storage technology on the capital cost of the subsystem. Figure 7.2 shows the analysis for the change in the subsystem cost of the ice and chilled water storage systems compared with the conventional system.

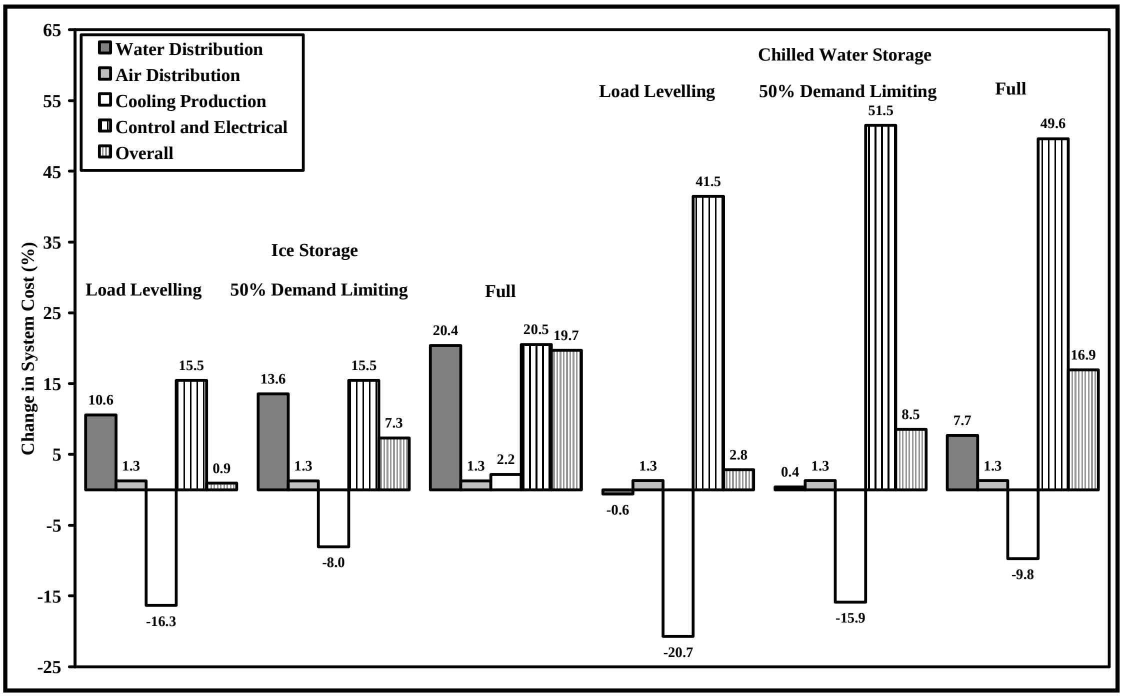

Figure 7.2 Change of subsystem cost of the ice and chilled water storage system compared with conventional system.

Figure 7.2 shows that the cost of the water distribution subsystem in the ice storage system is much higher than that of the conventional system. This increase in the cost resulted in the additional cost of pipes and valves associated with the ice tanks, and additional cost for the larger sizes of the chilled water pumps as well as the cost of the glycol solution, which was considered to be part of the water distribution subsystem. An air distribution subsystem is unlike other subsystems; the change in its costs for all AC systems is small since the duct and AHUs costs were approximately similar.

Similarly, the control and electrics costs also increase with the ice and chilled water storage systems as shown in Figure 7.2. It is clear that the change in the cost of the control and electrics especially with the chilled water storage systems increased from between 41.4% and 51.5% depending on the operating strategy. This increase in cost is a consequence of the addition of the extra control valves associated with the storage tank, temperature, and flow sensors; hence, the increased cost of the controllers, wiring and insulations.

Furthermore, Figure 7.2 illustrates that the use of cool thermal storage technology can reduce the cost of the cooling production subsystem (that is, the chiller cost); the reduction in the cost of the cooling production is more for the chilled water system than for the ice storage systems where the cost decreases by between 9.8% to 20.7% depending on the operating strategy of the system as shown in the figure. Moreover, the cost of the cooling production in the ice storage system decreases by 16.3% with load levelling and by 8% with demand limiting partial storage. However, for the full storage strategy, the cost increases slightly by about 2.2% compared with the conventional system.

In addition, it can be seen from the figure that the overall capital cost of the ice and of the chilled water storage systems are higher than those of the conventional system; the overall cost reaches the highest value in the ice storage system with a cost of 19.7% and in the chilled water storage system with a cost of 169% operating with full storage strategy, and the lowest value of just 0.9% in the ice and 2.8% in the chilled water storage operating with load levelling strategy. The resulting increase in the overall cost of the ice and chilled water storage systems was due mainly to the additional cost of the storage tanks, which ranges from between 6.3% and 14.3% of the total capital cost depending on the storage type and operating strategy (see Figure 7.1).

7.4 Connected power and water costs

Other additional costs that must be considered and added to the capital cost of the AC system in order to be included in the life cycle cost analysis are the cost of the connected power and water. The power connection cost is defined as the cost of the expected maximum power withdrawn from the utilities by the user at any time during the operation of the AC system, and it represents the actual cost of the power plants, electricity distribution lines, and transformers by the Ministry of Energy.

The actual cost of the connected power as estimated by the Ministry of Energy is 1383.6 US$ per kW_e. However, the Ministry charges the user about 173 US$ per $kW_{e,}$ which means the cost of the power connection is subsidised by 87.5% by the government. Likewise, the actual cost of the water production was estimated to be

2.213 US$ per m^3; however, the consumer pays only 0.6087 US$ per $m^{3,}$ hence, the cost of the water is subsidised by 72.5%. To account for the difference in the costs of power and water for the user and the nation, the life cycle cost was performed for two cases using the actual cost obtained by the government and the subsidised cost.

7.5 Operating and maintenance

The operation and maintenance costs can make a substantial impact on the overall annual recurring cost of the AC system, thereby dictating whether or not they are chosen. Therefore, an accurate estimation of the operating and maintenance cost is very important for determining the accurate life cycle cost of the AC system.

7.5.1 Operating

The operating costs are those incurred by the operation of the AC system. For Kuwait, they mainly include electricity and water costs; other costs such as operators, labour wages and spare parts were included with the maintenance cost and will be discussed in the next section. The annual electrical energy requirement for the operation of an AC system is an important technical and economic parameter for the cost effective selection of an AC system. The annual electrical energy was estimated as the summation of the hourly power demand over the complete running period of the AC system. The power demand as stated before is a function of the building total cooling demand and weather conditions.

Electrical operating costs for the AC systems are considered to comprise those for the chillers, AHUs and primary and secondary chilled water pumps. The annual electricity cost was calculated based on the estimated electricity cost by the government, which is about 0.069 US$ per kW_eh and the 90% subsidised cost for the user of about 0.007 US$ per kW_eh.

The water consumption in an air cooled system is usually minimal since the chilled water piping system is a closed system. The annual water consumption may result only from water leakage from the piping system and from partial water flushing during the annual maintenance of the AC system. Therefore, in the calculation for the amount of water consumed annually, it was assumed that the annual water consumption would be about 25% of the total system volume.

The cost of the water was calculated using the government figures of 2.214 US$ per m^3 and for the user of 0.609 US$ per m^3. It should be noted that the costs of the electricity and water actually vary with the price of fuel, and that the estimated costs of the electricity and water do not vary during the year and are estimated based on the fuel price of 20 US$ per barrel.

7.5.2 Maintenance

Maintenance costs depend on many parameters, such as labour rates, the experience of the workers, the age of the system, and the length of time of operation, and therefore are difficult to quantify (Aktacir et al. 2006). Chiller maintenance activities with ice and chilled water storage systems are the same as for a conventional system (Dorgan, 1994). However, in ice and chilled water systems, the chillers are smaller (except with an ice storage full strategy system), which should generally reduce the cost of the replacement parts and specifically reduce refrigerant replacement.

The addition of storage tanks in ice and chilled water systems is the most obvious difference from a conventional AC system, but incremental maintenance requirements are likely to be small. The water level in the storage tanks, control set points, valve operation and so on, should be checked periodically, and makeup water should be added if necessary.

For the ice storage systems, it is important to check the glycol concentration level on a regular basis, and it should be corrected as necessary together with the calibration and accurate operation of the ice inventory device (CIBSE, 1994). Furthermore, the maintenance of a chilled water storage system is similar the maintenance of a conventional system, expect for the larger volume and the need for additional considerations as detailed below (Dorgan, 1994).

a. Low circulation rates through stratified tanks make after the fact cleaning and flushing difficult.

b. The materials used in the tank construction or tank surface finishing need to be included in the corrosion review.

Based on the technical information, such as system description, operating procedures and seasonal start up and shut down provided to the contractor, the maintenance cost (including labour salary) was estimated to be 10.33 US$ per kW_t per year for the conventional system and 11.31 US$ per kW_t for the ice and chilled water storage systems. The maintenance costs of the ice and chilled water storage systems are slightly higher compared with those of a conventional system; this additional cost was added by the contractor to account for the larger water volume in the ice and chilled water storage systems where additional chemical treatment is required, along with more services of the control valves and sensors.

7.6 Life Cycle Cost

Generally, to calculate the life cycle cost for each AC system, all costs must be identified by year and by amount and this requires extensive calculations, especially when the study period is more than a few years long and is for annually recurring amounts, for which future costs must first be calculated to include changes in prices. However, Sieglinde (1996) provided a simplified formula, which can be relied on and therefore can be used to estimate the life cycle cost in present value; from this, different AC systems designs can be compared with the conventional system.

7.6.1 Life cycle cost analysis calculation

The following simplified formula was used to compute the life cycle cost for the conventional, ice and chilled water AC systems as stated by Sieglinde (1996)

$$\mathrm{LCC} = \mathrm{C} + \mathrm{Rep} + \mathrm{E} + \mathrm{W} + \mathrm{OMR} \tag{7.1}$$

The present value is defined as the time equivalent value of past, present or future cash flows as of the beginning of the base years (Sieglinde, 1996). For all expenses given in equation (7.1), it was determined using the following formula (Sieglinde, 1996)

$$\mathrm{PV} = \mathrm{F} \times \mathrm{UPV} \tag{7.2}$$

The uniform present value factor UPV in equation (7.2) was calculated as follows (Sieglinde, 1996)

$$\mathrm{UPV} = \frac{(1-e)}{(i'-e)}\left[1-\left(\frac{1+e}{1+i'}\right)^{n}\right] \tag{7.3}$$

By using the uniform present value factors UPV defined in equation (7.3), the life cycle cost LCC was calculated without first computing the future annual amount including the price escalation of each annually recurring cost over the entire life of the system, summing all those costs by year and discounting them to present value PV. Instead, the life cycle cost was estimated by calculating only the annual amount in base year dollars (that is, a one time amount) and identifying the uniform present value factor UPV.

The effective interest rate i' in equation (7.3), sometimes called the real rate, is defined as the rate of interest reflecting the investor’s time value of money (Kirk, 1995). Basically, it is the interest rate that would make an investor indifferent as to whether he received a payment now or a greater payment at some time in the future. The effective interest rate i' was calculated by ASHRAE (ASHRAE, 1999a)

$$i' = \frac{i-j}{1+j} \tag{7.4}$$

The effective interest rate i' in equation (7.3) was determined by taking account of both the interest rate i and the inflation rate j. The inflation rate reflects the rise in the real costs of a commodity over time (ASHRAE, 1999a).

7.6.2 Financial input data

As stated earlier, the evaluation technique used was the present worth method, which compares the equivalent cash needed on hand to own and operate a system over an entire life of the system. This method produced a single investment value in dollars at the beginning of the system.

The financial analysis was performed based on a twenty-five year period, which is the expected life time for an air cooled system. Furthermore, the interest rate i' was taken as 5.26% and was obtained by taking the average interest rates of the previous four quarters, the last quarter of year 2005 and three quarters of year 2006 (CBK, 2006a). The inflation rate was estimated for the maintenance at 2.12% and was calculated using the increase in the customer price index between the second quarter of year 2005 and the second quarter of year 2006 (CBK, 2006b). However, the inflation rate for the unit cost of electrical energy and water was assumed to be zero. Moreover, the escalation rate, e, of the maintenance was assumed equal to 3% to account for the increase in the operator's salary and the increase in the cost of the number of spare parts as the system becomes older.

7.6.3 Financial analysis

Based on the above input data, the total life cycle cost LCC in present value for each AC system design was calculated. Table 7.1 shows a sample of the results summary for the calculation of the life cycle cost LCC for the conventional AC system that was calculated based on the power connection and electrical energy costs estimated by the government.

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	637,838	Base Date	Already in Present Value	637,838
Maintenance at 10.33 ($ per $kW_{t)}$	4,359	25	32	139,748
Annual Electricity Consumption 0.069 ($ per kW_eh)	37,114	Annual	17	640,499
Annual Water Consumption 2.214 ($ per m^3)	2	Annual	17	37
Total Life Cycle				1,418,121

Table 7.1 Data summary of the life cycle cost LCC for conventional AC system

In Table 7.1, the present value PV for each expense was obtained by multiplying the second column (that is, base date cost) by the fourth column (that is, uniform present value factor UPV), and then the total life cycle cost LCC in present value was obtained by summing all the expenses in the present value.

Moreover, the life cycle cost of US$ 1.4 thousand in Table 7.1 for the conventional AC system serves as a baseline against which the life cycle cost of the ice and chilled water storage AC systems will be compared based on the estimated energy consumption and power connection costs for the government and the user. Figure 7.3 shows the change in the present value for the expenses except the water consumption and total life cycle cost of the ice and chilled water storage systems compared with the conventional system. The water consumption is not included in the analysis because it has an insignificant effect on the overall life cycle cost for all AC systems. However, it should be noted here that with the chilled water storage system, the water consumption is much more than it is for other systems because of the presence of large water tanks.

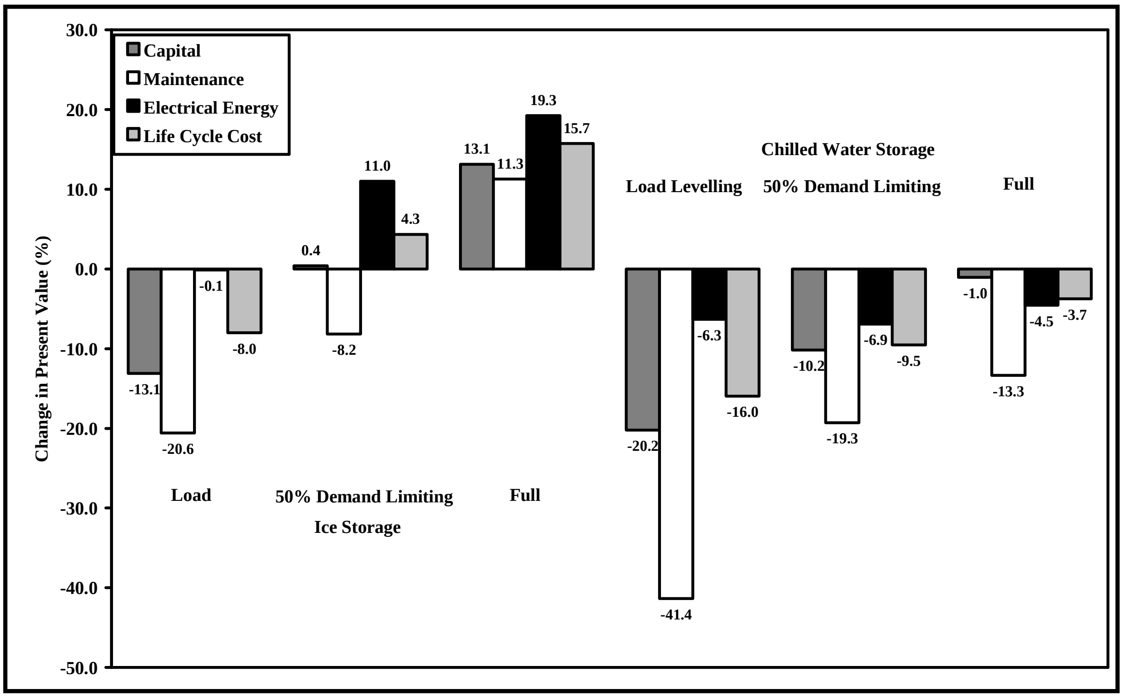

Figure 7.3 Change in present values based on government costs of ice and chilled water AC systems compared with the conventional system.

The result in the figure was calculated based on the estimated energy consumption and power connection for the government. The results indicate that all AC systems except ice storage system operating with 50% demand limiting and full strategies have a lower life cycle cost compared with the conventional system at an expected system life time of twenty five years; therefore, these systems can be considered to be cost effective. Furthermore, a chilled water storage system operating with a load levelling strategy has the lowest life cycle cost of 16%, and this is mainly due to the much lower capital cost of 20.2% and maintenance cost of 41.4% resulting from much lower power connected costs and hence a much lower capital cost (see Appendix H).

Furthermore, it can be seen from the figure that all the expenses in the ice storage system with load levelling are lower than for the conventional system; as a consequence, the life cycle cost decreases to about 8.0%. Moreover, the life cycle cost in an ice storage operating system is higher than that of the conventional system by 4.3% for 50% demand limiting strategy and by 15.7% for full strategy; the higher costs are mostly due to the higher energy consumption cost in these systems. The case of chilled water storage systems is different. The change in the life cycle cost for all strategies is lower by between 3.7% and 16% depending on the operating strategies. It can also be observed that all systems have lower expenses compared with the conventional system.

The change in the present values are also plotted for the expenses based on the subsidised costs for the energy consumption and power connection (that is, for the user) as shown in Figure 7.4. It is clear that the change in the present values of the energy consumption and maintenance are the same as those given in Figure 7.3; however, the calculated present value in US$ for the user is much lower than the energy consumption calculated for the government because the energy cost is subsidized. Furthermore, it can be observed from Figure 7.4 that all the capital costs in the LCC increases when compared with those given in Figure 7.3. The reduction in the capital costs was due to a much lower power connection cost provided to the user by the government.

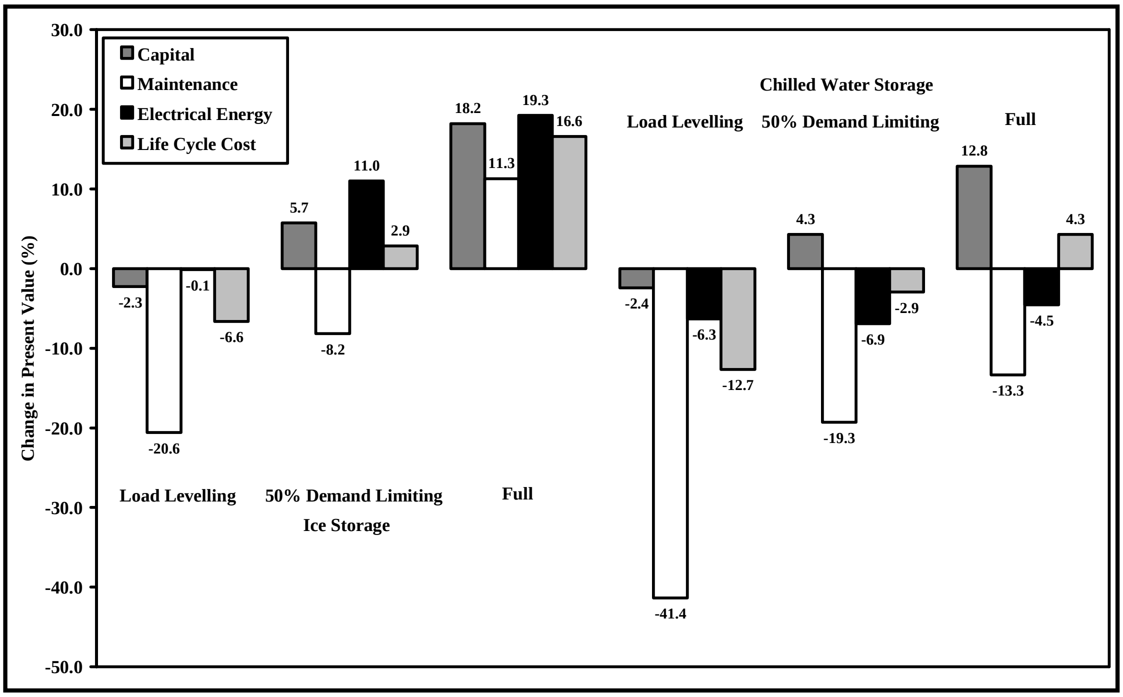

Figure 7.4 Change in present values based on user costs of ice and chilled water AC systems compared with the conventional system.

Moreover, ice storage with load levelling and chilled water storage with load levelling and 50% demand limiting strategies have lower life cycle costs compared with the conventional system; therefore, they can only be considered cost effective for the user. Again, a chilled water storage system with a load levelling operating strategy has the lowest life cycle cost of 12.7%, an ice storage system operating with load levelling strategy is the second lowest with a cost of 6.7%, and for a chilled water storage system operating with 50% demand limiting strategy, the reduction in the proportion of life cycle cost is about 2.9%.

Furthermore, it can be seen from the figure that change in the present value of the capital cost of the chilled water storage system operating with full strategy increases by 12.8%; however, for the government it is reduced by 1.0% as shown in Figure 7.3. As a result, the life cycle cost increases to about 4.3% for the user. This means that a full storage strategy in the chilled water storage system is attractive only for the government, but not for the user. In addition, Figure 7.3 and Figure 7.4 show that the use of an ice storage system operating with 50% demand limiting and full strategies is not cost effective and, therefore, should not be implemented in Kuwait.

7.7 Conclusions

In this chapter, the present value of the life cycle cost was determined for conventional, ice and chilled water storage AC systems based on the expected system life of twenty-five years. Each expense such as capital, maintenance, and operating costs that was involved in the calculation of the life cycle cost was estimated. The capital costs of the AC systems were obtained mostly from the supply and contract agencies in Kuwait. The maintenance costs were estimated in US$ per Kilowatt cooling based on the system specifications and factors that directly affect the maintenance costs. Furthermore, the operation costs including electrical energy and water consumption were calculated based on the actual estimated cost by the government and the subsidised cost for the user.

The capital costs of the AC systems and their subsystems excluding the power connection and water costs were analysed. It was found that the cooling production subsystems have the largest proportion to the overall capital costs of AC systems compared with other subsystems. This ranged from between 39.5% and 51.2%. Furthermore, storage tanks were considered an additional subsystem in ice and chilled water storage systems; their proportion ranged from between 6.3% and 14.3% depending on the type of storage technology and the operating strategy. Moreover, by further examining the change in the capital cost of the ice and chilled water systems compared with the conventional system, it was found that the costs of the cooling production subsystems were lower by between 8.0% and 20.7% than the conventional system. However, in the ice storage system operating with full strategy the cost was slightly higher by 2.2%. Other subsystems in ice and chilled water systems were found to be unlike the cooling production subsystem; their costs were higher than were those of the conventional systems and for that reason the overall capital costs of the ice and chilled water systems were higher than those of the conventional system by between 0.9% and 19.7%.

In addition, based on the analysis of the life cycle cost of the systems, a chilled water storage system operating with a load levelling strategy was found to be the most cost effective design for both the government and user. Because of a much smaller system size in this design, the calculated present value of the capital (that is, AC and power connection costs) and maintenance costs were much lower than were those for the conventional system. Furthermore, in ice storage systems, it was observed that load levelling strategy partial storage had a lower life cycle cost of only 8.3% for the government, and 6.7% for the user when compared with the conventional system.

For the government, the life cycle costs for all chilled water storage systems were found to be lower by between 3.7% and 16% than were those for the conventional systems. However, for the user, the life cycle costs were lower by 12.7% with load levelling and 2.9% with 50% demand limiting strategies and for the full strategy the life cycle costs increased by 4.3%. It should be noted that although the life cycle cost for the load levelling strategy is lower in ice and storage systems, the reduction in the maximum power demand is lower than in other strategies. So depending on the building's owner, the choice can be made between an AC system with a lower capital cost and less reduction in the maximum power demand or a higher capital cost system with a greater reduction in the maximum power demand.

A sensitivity analysis has been performed to identify the parameters having the highest influence on the estimation of the life cycle cost. The parameters such as cost of electricity, inflation rate, interest rate, and plant lifetime were varied from their default value by considering possible variation in their range in the context of Kuwait. The analysis showed that the cost of electricity had the largest impact on the life cycle cost because of large variation of the price of oil in the last ten years. The price of oil in ten years varied from average 10 US$ per Barrel in 1997 up to 60 US$ per Barrel in 2006. The electricity cost estimated by the government was based on the cost of oil at 20 US$ per Barrel.

The cost of electricity was recalculated based on different cost of oil (e.g. 10, 35 and 60 US$ per Barrel), therefore, the parameter was varied between -40%, 105% and 240%. The results showed that the life cycle cost was highly effected by the large variation in the oil price, with a -40% decrease in the electricity price the percentage decrease in the life cycle cost was between -20.2% and -17.9%. With a 150% increase in the price of oil the life cycle cost increased by between 22.4% and 25.2% and with a 240% increase in the oil price life cycle cost increased between 62.7% and 85.0%.

Based on the available data for the inflation and interest rates from the Central Bank of Kuwait (CBK), the effect of a variation of ±20% for the inflation rate and ±25% in the interest rate was considered. The impact of the interest rate on the life cycle cost of the AC systems ranged from between -9.5% and -8.9% depending of the AC system design when the interest rate was reduced to by 25% and ranged from between 7.2% and 7.7% when the interest rate was increased by 25%. Furthermore, the impact of the inflation rate was small when compared with other parameters, life cycle costs varying by about 2.7% to -2.8% when the inflation varied by ±20%.

The plant lifetime was also varied from the default value of 25 years. Due to leak of acceptable data, it is not possible to quantify the extent of variation. So, a typical assumption of ±20% was made to see its influence on the life cycle cost estimate. The impact of the plant lifetime ranged from between -9.1% and -8.6% depending on the AC system design when the plant lifetime was reduced by 20% and ranged from between 7.9% and 8.3% when the plant lifetime was increased by 20%.

Chapter 8

Conclusions and Recommendations

8.1 Conclusions

In Kuwait, the demand for electricity has recently become a topic of major interest and prominence. Electricity is a highly capital-intensive sector, which requires relatively long lead times for construction and development. The gap between supply and demand is shrinking and new projects require huge investment, the financing of which faces a dilemma in the light of high fluctuations in oil revenues.

In summer 2006, Kuwait faced a series of power shortages emphasizing the need for the urgent commissioning of power generation projects. It has been shown that the demand for electricity is growing at an average of 6.2% per year, encouraged by government subsidies and driven by the rapid and continual expansion in building construction, urban development, and the heavy reliance on AC systems for the cooling of buildings (see Section 2.3). It has been estimated that AC alone consumes about 39% of the total electricity generated per year and 45% of the exported electrical energy per year from the power stations (see Section 2.6.1). In addition, it has been determined that the proportion of AC systems to electricity generation during the peak demand period is about 62% (see Section 2.6.3). Consequently, the development and utilisation of energy efficient systems strategies has become of national interest. Within the AC system, it has been found that air-cooled chillers consume about 76% of the electricity at the maximum power demand and, therefore, have the largest power consumption compared with other components (see Section 2.7.1).

The cool thermal storage system is one of the available techniques for reducing the growth of electricity demand by simply shifting the energy consumption of the AC systems from day time, when the demand of the electricity is high, to night time when the demand of electricity is low. By applying this simple technique, not only the power demand in the day time can be reduced but also the size of the AC systems can be reduced and hence the capital, maintenance and possibly the operating costs. The main objective of this thesis was to evaluate cool thermal storage strategies in Kuwait in terms of their power reduction, energy consumption and economics.

It is well known from the literatures and case studies of the existing operation of cool thermal storage systems that the power demand can be reduced; however, there are other factors that must be considered when using such systems in Kuwait, including the overall energy consumption and economics. In some countries, where the cost of electricity varies during the day, cool thermal storage is considered a cost saving technology and not an energy saving technology, which means it saves operation costs but not energy consumption. In Kuwait, the energy costs have a linear pattern and are highly subsidised by the government; therefore, applying cool thermal storage must be investigated in detail by taking account of the different available types of cool thermal storage systems, operating strategies, and controls.

In this thesis, two types of cool thermal storage systems were found to be suitable for Kuwait, namely, internal ice-on-coil ice storage, and chilled water storage systems; therefore, their performances were examined and compared with that of the conventional system. In order to perform the energy analysis and estimate the cost of the systems, first the building energy simulation ESP-r was used to calculate the hourly cooling load of the building, then each component including chillers, storage tanks, AHUs and pumps in the AC systems were sized; the heat gains by the auxiliary systems were then determined and added to the building cooling load. Thus, the total cooling load of the building was developed for all AC systems (see Chapter 5).

In addition, for each AC system, the input power and energy consumptions for the design day and for the whole year were calculated and their performances were analysed (see Chapter 6). For the design day, it was found that the chillers in all systems used the largest proportion of the energy consumption, from between 75.8% and 78.3% compared with other components. They also had the largest power consumption, excluding full strategies systems at the maximum demand time, from between 72.2% and 85.5%.

Moreover, the use of ice storage AC systems reduced the maximum power demand requirement in the day time by between 19.4% and 83.8% compared with the conventional system. However, the energy consumption increased by between 6.1% and 24.3% depending on the design operating strategies. With the chilled water storage systems, the power demand at maximum cooling load decreased by between 36.7% and 87.5% and the energy consumption decreased by between 5.5% and 7.2% depending on the selected strategy.

In terms of the annual performance of the AC systems, it was concluded that the annual energy consumption of the ice storage systems operating with 50% demand limiting strategies increased by 11.0% and for full strategies by 19.3%. However, with a load levelling strategy, the energy reduced by about 0.9% compared with the conventional AC system.

With the chilled water storage AC system, the maximum power demand in the day time decreased by between 36.7% and 87.5% depending on the design of the operating strategies. The chilled water storage system operating with full strategy was found to have the largest reduction in the maximum power demand of 87.5% compared with all systems. This reduction was established as a result of switching off the chiller and primary pump completely during the day time.

Furthermore, the results for the annual performance of the chilled water storage system were found to be better than the ice storage and conventional systems. Compared with the conventional system, the annual energy consumption was improved by between 4.5% and 6.3% depending on the operating strategy. This reduction in the energy was brought about mainly by a saving in energy of the chilled water pumps and chiller.

Given the conclusions outlined above, the chilled water storage system is recommended as the best option for cooling of typical medium size buildings in Kuwait since both the maximum power and annual energy consumptions were reduced. In addition, the building owner has the choice of which strategy his AC systems can be designed. The largest reduction in power can be achieved with a full strategy, the smallest with a load levelling strategy and somewhere in between can be achieved by a 50% demand liming operating strategy. However, with chilled water storage systems, because of the large volume of the storage tanks associated with these systems, the storage tanks require additional space more than any other AC systems. In new constructions, the tanks can be constructed and located, for example, in the basement of the building, or could be part of the building or buried, and this could help to reduce the cost of the tank and save space.

Economic analyses were also carried out for each AC system to assess the life cycle cost based on the actual estimated costs by the government for the water and energy consumptions and power connection and on the subsidised costs to the user. The capital, maintenance, and operation costs were the main imported evaluated term in the life cycle cost of the systems. Based on the analysis of the capital cost of the AC systems, it was observed that the largest proportion of the overall capital costs of the AC systems was contributed by the cooling production subsystem and ranged from between 39% and 51.2%. With ice and chilled water AC systems, the costs of the ice and water tanks were from between 6.3% and 14.3% of the overall system cost. Generally, the overall capital costs of the ice and chilled water storage systems were found to be from between 0.9% and 19.7% higher than for the conventional system.

From the results analysis of the estimated life cycle cost for the AC systems, the following conclusions are drawn:

a. For the government, the estimated life cycle costs of the chilled water storage were all lower than for the conventional systems by between 3.7% and 16.0%; however, for the user only costs load levelling and 50% demand limiting strategies were found to be lower.
b. In the ice storage systems, the life cycle cost of the load levelling strategy was the only one that was lower than the conventional system, by 8.3% based on the government estimated cost and 6.7% based on subsidised costs.

Based on the above conclusions, the chilled water thermal storage AC system operating with a load levelling strategy represents the most cost-effective method for a typical medium size building for both the government and user; therefore, this is the recommended option for the conditions in Kuwaiti. With this storage technology and strategy it was observed that the size of the AC system was smallest compared to other storage systems; therefore, there was a reduction in the capital, maintenance and more importantly the power connection costs of the system.

Although this option has the lowest energy consumptions, the reduction in the power demand is less than that of other strategies of the chilled water storage system. For the government, if the higher reduction in the power demand is required, 50% demand limiting or full strategies is recommended, and for the user only a 50% demand limiting strategy is recommended.

The results from this thesis can not generalized for all types and sizes of buildings however, the method of sizing of each component of the AC systems can be followed. In addition, the procedures of conducting the energy and economic analysis can be used.

8.2 Recommendations for future work

To improve further the operation and efficiency of the ice storage system in order to reduce the energy consumption it is recommended that building owners

a. Use chillers with higher coefficients of performance.

b. Use more than one chiller in the system; one chiller can be used for charging the ice tank and another chiller can be used for directly cooling the building, although this suggestion might increase the capital cost of the system for the clinic but it may improve the energy consumption.

c. Since with the ice storage system a low chilled water temperature can be produced, the air distribution system can be designed with a much colder air distribution. This will help to reduce the energy consumption, power connection and costs of the AHUs.

It has been shown in this research that the chilled water storage system was economically attractive for typical medium size buildings such as the clinic and, in fact, the chilled water storage system is increasingly economical with a larger buildings; hence, additional study is required to investigate this issue.

This research work was conducted to assess cool thermal storage technologies operating with an air-cooled AC system; further research can be done to assess this technology with a water-cooled AC system. With a water-cooled system, the cooling production subsystem can be further reduced with cool thermal storage.

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Appendix A

Psychrometric Analysis of the Air Distribution System

A.1 Notation

$\dot{V}_a$	Volumetric flow rate of the supply air	(m^3s^{-1})
$\dot{Q}_l$	Latent heat gain of zone	(W_t)
$\dot{Q}_m$	Heat equivalent of the fan motor operation	(W_t)
$\dot{Q}_s$	Sensible heat gain of the zone	(W)
c_p	Specific heat of supply air	($Jkg^{-1}°K^{-1}$)
g	Moisture content	(kg water per kg dry air)
g_{a1}	Moisture content at air stream 1	(kg water per kg dry air)
g_{a2}	Moisture content at air stream 2	(kg water per kg dry air)
g_{a3}	Moisture content at mixing	(kg water per kg dry air)
g'_{ss}	Moisture content of saturated air at a temperature t´	(kg water per kg dry air)
g_{ss}	Moisture content of saturated air	(kg water per kg dry air)
g_Z	Moisture content of zone	(kg water per kg dry air)
h	Specific enthalpy of moist air	($kJkg^{-1}$)
h_a	Specific enthalpy of dry air	($kJkg^{-1}$)
h_{fg}	Latent heat of evaporation	(Jkg^{-1})
h_g	Specific enthalpy of water vapour	($kJkg^{-1}$)
h_1	Enthalpy of moist air at stream 1	($kJkg^{-1}$ of dry air)
h_2	Enthalpy of moist air at stream 2	($kJkg^{-1}$ of dry air)
h_3	Enthalpy of moist air at mixing	($kJkg^{-1}$ of dry air)
m_a	Mass of dry air	(kg)
$\dot{m}_{a1}$	Mass flow rate of dry air at stream 1	(kgs^{-1})
$\dot{m}_{a2}$	Mass flow rate of dry air at stream 2	(kgs^{-1})
$\dot{m}_{a3}$	Mass flow rate of dry air at mixing	(kgs^{-1})
p_{at}	Atmospheric pressure	(Pa)
p_s	Pressure of water vapour	(Pa)
p_{ss}	Saturation vapour pressure	(Pa)
t	Absolute temperature	(K = °C + 273.15)
T	Temperature	(°C)

t'	Wet-bulb temperature	(°C)
t_d	Dew-point temperature	(°C)
T_m	Temperature of the moist air	(°C)
T_s	Dry bulb temperature of supply air	(°C)
T_z	Dry bulb temperature of the zone	(°C)
v	Specific volume	(m^3kg^{-1} dry air)
V	Total volume of the mixture	(m^3)
α	$\ln(p_s)$	-
Δt_m,	Temperature rise across fan motor	(°K)
ρ	Density of air	(kgm^{-3})
ρ_a	Density of supply air	(kgm^{-3})
φ	Relative humidity	-
μ	Percentage saturation	-

A.2 Calculation of psychrometric properties of moist air

In order to estimate correctly the heat gains by the AHUs' fan motors and supply and return ducts, and thus estimate the cooling load on the cooling coil, a psychrometric analysis was performed for the AC processes at each state in the air distribution system and within the zone.

In this section, the formulae for calculating the psychrometric properties of moist air that were used to describe the state of the AC processes for the zones 1 to 6 in the clinic are introduced. The formulae that are used in this section were taken from the ASHRAE handbook (Jones, 1994; ASHRAE, 2001).

A.2.1 Saturation vapour pressure

The calculation of the saturation vapour pressure is required to determine a number of moist air properties, mainly the percentage saturation. The saturation pressure for the temperature range of 0 to 200° C is given by

$$p_{ss} = \mathrm{Exp}\left[C_1 t^{-1} + C_2 + C_3 t + C_4 t^2 + C_5 t^3 + C_6 \ln(t)\right] \tag{A.1}$$

where

$C_1 = -5.8002206\times10^3$

$C_2 = 1.3914993$

$C_3 = -4.8640239\times10^{-2}$

$C_4 = 4.1764768\times10^{-5}$

$C_5 = -1.4452093\times10^{-8}$

$C_6 = 6.5459673$

A.2.2 Moisture content of dray and saturated air

The moisture content of a given sample of moist air is defined as the ratio of the mass of water vapour to the mass of dry air contained in the mixture of the moist air. The moisture content can be calculated as

$$g = 0.62198\frac{p_s}{(p_{at} - p_s)} \tag{A.2}$$

For moisture content of the saturated air, equation (A.2) becomes

$$g_{ss} = 0.62198\frac{p_{ss}}{(p_{at} - p_{ss})} \tag{A.3}$$

A.2.3 Percentage saturation

Percentage saturation is defined as the ratio of moisture content of the moist air to the moisture content of the saturated air at the same temperature and pressure. This is expressed as

$$\mu = \frac{g}{g_{ss}} \tag{A.4}$$

The percentage saturation μ is also know as the degree of saturation.

A.2.4 Relative humidity

The relative humidity is defined as the ratio of the partial pressure of water vapour in the moist air to the partial pressure of saturated water vapour in the air at the same temperature, i.e.

$$\varphi = \frac{p_s}{p_{ss}} \tag{A.5}$$

Alternatively, the relative humidity φ can be determined in terms of saturated vapour pressure p_{ss} and percentage saturation μ by the following equation

$$\varphi = \frac{\mu}{1-(1-\mu)(p_{ss}/p_{at})} \tag{A.6}$$

Equation (A.5) can also be used to determine the pressure of water vapour p_s, if the relative humidity φ and saturated vapour pressure p_{ss} are known, i.e.

$$p_s = \varphi \times p_{ss} \tag{A.7}$$

A.2.5 Specific volume

Specific volume is defined as the volume of the dry air mixed with water vapour when the mass of the dry air is equal to 1kg, i.e.

$$v = \frac{V}{m_a} \tag{A.8}$$

Another relation that can be used to determine the specific volume v, in terms of atmospheric pressure p_{at}, and moisture content g, of the moist air is given by

$$v = 0.2871\frac{(T + 273.15)(1+1.6078g)}{p_{at}} \tag{A.9}$$

A.2.6 Dew point temperature

The dew point temperature is the temperature of saturated air that has the same vapour pressure as the moist air under consideration. A dew point temperature ranging from 0 to 93° C can calculated from,

$$t_d = d_1 + d_2\alpha + d_3\alpha^2 + d_4\alpha^3 + d_5(1000p_s)^{0.1984} \tag{A.10}$$

where

d_1 = 6.54

d_2 = 14.526

d_3 = 0.7389

d_4 = 0.09486

d_5 = 0.4569

A.2.7 Wet bulb temperature

The wet bulb temperature is the temperature indicated on a thermometer, the bulb of which has been wrapped round with a wick that has been moistened in water. The wet-bulb temperature can be calculated by solving the following equation numerically

$$g(2501+1.805T - 4.186t') - (2501 - 2.381t')g'_{ss}(p_{ss}(t')) + 1.006(T - t') = 0 \tag{A.11}$$

and

$$g'_{ss}(p(t')) = 0.62198 \frac{p_{ss}(t')}{(p_{at} - p_{ss}(t'))} \tag{A.12}$$

The function $p_{ss}(t')$ in equation (A.12) can be obtained by substituting the wet bulb temperature t' instead of dry bulb T in equation (A.1). If equation (A.1) is substituted into equation (A.12), and then equation (A.12) is substituted into equation (A.11), the only unknown in equation (A.11) will be the wet bulb temperature t' which can be determined by solving the equation numerically.

A.2.8 Specific enthalpy of the moist air

The enthalpy of a sample of moist air is defined by the following equation

$$h = h_a + gh_g \tag{A.13}$$

In equation (A.13), the specific enthalpy of the dry air h_a, and water vapour h_g are respectively defined as

$$h_a = 1.007T_m - 0.026 \tag{A.14}$$

and

$$h_g = 2501 + 1.84T_m \tag{A.15}$$

By substituting equations (A.14) and (A.15) into equation (A.13), the specific enthalpy of the moist air can be determined as

$$h = 1.00T - 0.026 + g(2501 + 1.84T_m) \tag{A.16}$$

A.2.9 Mixture of two air streams

The mixing of two air streams is encountered in many AC processes, such as the mixing of return air with fresh air and bypassing mixing. When two air streams mix adiabatically, such as when the fresh air mixes with the return air, the conservation of the mass and energy can be applied to determine the air properties at the mixing condition. From the conservation of mass, the mass balance of the dry air and the associated water vapour are respectively given by,

$$\dot{m}_{a1} + \dot{m}_{a2} = \dot{m}_{a3} \tag{A.17}$$

and

$$g_1\dot{m}_{a1} + g_2\dot{m}_{a2} = g_3\dot{m}_{a3} \tag{A.18}$$

Solving equations (A.17) and (A.11) for the moisture content at the mixing state i.e.

$$g_3 = \frac{g_1\dot{m}_{a1} + g_2\dot{m}_{a2}}{(\dot{m}_{a1} + \dot{m}_{a2})} \tag{A.19}$$

and from the conservation of energy

$$\dot{m}_{a1}h_1 + \dot{m}_{a2}h_2 = \dot{m}_{a3}h_3 \tag{A.20}$$

From equation (A.17), substitute $\dot{m}_{a3}$ into equation (A.20), and solve for the specific enthalpy h_3 at mixing state i.e.

$$h_3 = \frac{\dot{m}_{a1}h_3 + \dot{m}_{a2}h_3}{(\dot{m}_{a1} + \dot{m}_{a2})} \tag{A.21}$$

It is also important to calculate the dry bulb temperature at mixing, so other psychrometric properties of the air can be determined. The moisture content g_3 and specific enthalpy h_3 at state 3 can be calculated from equations (A.19) and (A.14) respectively, and equation (5.16) of the specific enthalpy h can be used to calculate the dry bulb temperature T_3 i.e.

$$T_3 = \frac{h_3 - 2501g_3 + 0.026}{(1.007 + 1.84g_3)} \tag{A.22}$$

A.3 Numerical calculation of moist air properties

This section presents the steps that were followed to calculate the moist air properties at different states in the air distribution system using the formulae presented in section A.1. The states in the air distribution system are shown in Figure A.1, and denoted as point R which represents a state point in the return air from the zone, S which represents the state of the supply air to the zone; M which represents the off-coil mixing condition; and C which represents the on-coil state. Other state points in which the psychometric properties of the air were determined are point Z which represents the condition at the zone in the building and point O which is the outside design condition.

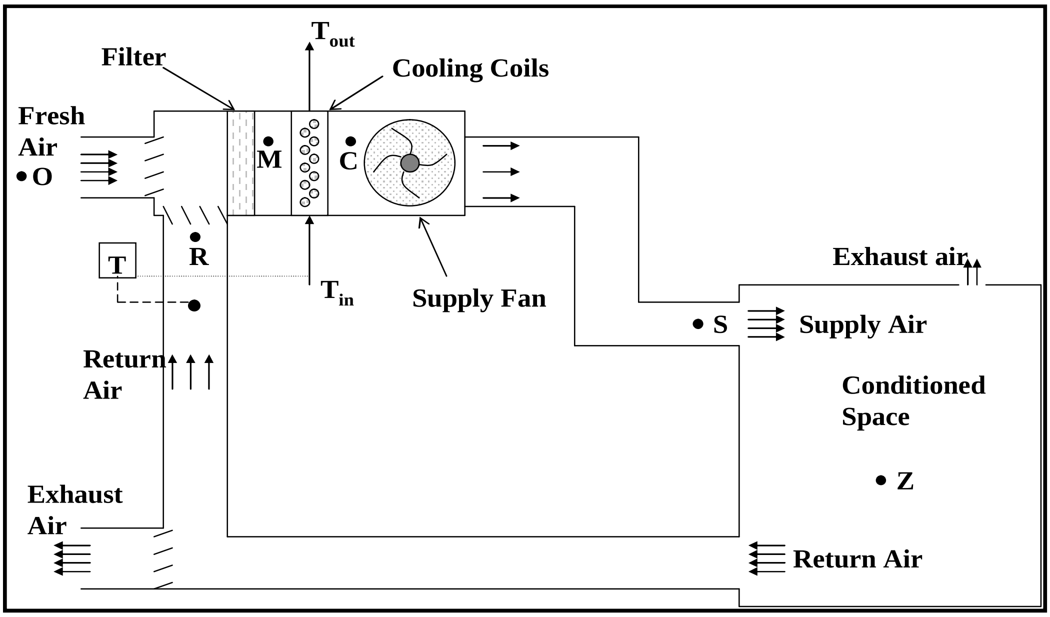

Figure A.1 Different states in the air distribution system.

A.3.1 Supply condition to the zone

The volumetric flow rate $\dot{V}_a$ of the supply air was determined by

$$\dot{V}_a = \frac{\dot{Q}_s}{\rho c_p (T_z - T_s)} \tag{A.23}$$

The sensible heat gain $\dot{Q}_s$ has already been determined from the building energy simulation program ESP-r for each zone in the clinic building and it is equal to the maximum building load (excluding ventilation load); the zone temperature T_Z was 24° C and the dry bulb temperature T_s of the supply air was assumed. Since the supply temperature T_S was lower than the dry bulb temperature in the zone T_Z by 8° C to 11° C (Jones, 1994), an average value was taken of this temperature range, which gives a supply temperature T_S of (T_Z – 9.5)° C.

The moisture content g, at the supply condition was obtained by

$$g = g_Z - \frac{v\dot{Q}_l}{\dot{V}_a \rho h_{fg}} \tag{A.24}$$

In equation (A.24), the latent heat gain $\dot{Q}_l$ in the zone was determined from the building energy simulation program ESP-r; the specific volume v was evaluated at temperature of 20° C and relative humidity of 50%; the moisture content g_Z of the zone was determined from the condition at the zone (for example, 24° C dry bulb temperature and 50% relative humidity); and the latent heat of evaporation h_{fg} was 2454 kJkg^{-1}, at 20° C.

Having obtained the moisture content g other properties of the supply air were obtained as follows

a. Substitute T_S, p_{at} and g into equation (A.9) to obtain specific volume v.

b. Substitute g'_{ss}, and T into equation (A.11), and solve the equation numerically for the wet bulb temperature t´.

c. Substitute T_S into equations (A.14) and (A.15) to obtain the specific enthalpy of dry air h_a and water vapour h_g respectively.

d. Substitute g, h_a and h_g into equation (A.13) to obtain the specific enthalpy of the moist air h.

A.3.2 Condition inside the zone

Since the dry bulb T_Z and relative humidity φ at the zone are known the psychrometric properties of the moist air in the zone were determined by the following steps

a. Substitute T_z into equation (A.1) to obtain the pressure of the saturation vapour, p_{ss}.

b. Substitute p_{ss} and φ into equation (A.7) to obtain the pressure of the water vapour, p_s.

c. Substitute p_s and p_{at} into equation (A.2) to obtain the moisture content, g.

d. Substitute p_{ss} and p_{at} into equation (A.3) to obtain the moisture content of the saturated air g_{ss}.

e. Substitute g and g_{ss} into equation (A.4) to obtain the percentage saturation μ.

f. Substitute T_z, p_{at} and g into equation (A.9) to obtain the specific volume v.

g. Substitute p_s into equation (A.10) to obtain the dew point temperature at the zone t_d.

h. Substitute g'_{ss} and T_Z into equation (A.11), and solve the equation numerically for the wet bulb temperature t´.

i. Substitute T_z into equations (A.14) and (A.15) to obtain the specific enthalpy of dry air h_a and water vapour h_g respectively.

j. Substitute g, h_a and h_g into equation (A.13) to obtain the total specific enthalpy of the moist air h.

A.3.3 Ambient condition

The outside summer design condition for the costal area in Kuwait is 47.4° C dry bulb, and 27.1° C wet bulb (Shaban, 2000). This design condition is based on the hourly ambient temperatures data recorded at the Kuwait Institute for Scientific Research (KISR), and is determined in terms of their 1% frequency of occurrence in the months of July and August in Kuwait. The frequency value represents the duration for which the temperature is equal to, or exceeds by one percent the design temperature during the months of July and August. Since the only data available for calculating the psychrometric properties of the air are the design ambient dry temperatures, T_O and wet bulb t´ temperatures, so the following steps were followed to determine other properties

a. Substitute t´ into equation (A.1) to obtain the pressure of the saturation vapour p_{ss}.

b. Substitute p_{at} and p_{ss} into equation (A.3) to obtain the moisture content of the saturated air g'_{ss} at temperature t´.

c. Substitute T_O, t´ and g'_{ss} into equation (A.11) to obtain the moisture content g.

d. Substitute T_O into equation (A.1) to obtain the pressure of the saturation vapour, p_{ss}.

e. Substitute p_{ss} obtained in step 4, and p_{at} into equation (A.3) to obtain the moisture content of saturated air g_{ss}.

f. Substitute g_{ss} obtained from step 5, and g into equation (A.4) to obtain the percentage saturation, μ.

g. Substitute p_{ss} obtained in step 4, p_{at} and μ, into equation (A.6) to obtain the relative humidity φ.

h. Substitute g, T_O and P_{at} into equations (A.9) to obtain the specific volume v.

i. Substitute g and p_{at} into equation (A.2) to obtain the pressure of the water vapour p_s.

j. Substitute p_s into equation (A.10) to obtain the dew point temperature t_d.

k. Substitute T_O into equations (A.14) and (A.15) to obtain the specific enthalpy of dry air h_a and water vapour h_g respectively.

l. Substitute g, h_a and h_g into equation (A.13) to obtain the total specific enthalpy of the moist air h.

A.3.4 Return condition (before air mixture)

Since there is no water vapour added to the return air when the air returns from the zone to the AHU, therefore the moisture content of the air g, in the return air duct just before the mixing, remains the same as the moisture content of the air in the zone. The dry bulb temperature of the return air T_R is equal to the dry bulb temperature of the zone T_Z plus the rise in the air temperature in the duct of the return air, which was assumed to be 0.15° C (El-Amer, 2005). The specific enthalpy of the moist return air was calculated using the following steps,

a. Substitute T_R into equations (A.14) and (A.15) to obtain the specific enthalpy of dry air h_a and water vapour h_g respectively.

b. Substitute g, h_a and h_g into equation (A.13) to obtain the total specific enthalpy of the moist air h.

A.3.5 Mixing (on-coil) condition

Having obtained the psychrometric properties of the air at the ambient and return points, equations (A.19), (A.21) and (A.22) were used to calculate the moisture content g specific enthalpy h, and the dry bulb temperature T_M respectively at mixture state (i.e. on-coil condition). Other air properties were calculated as follows,

a. Substitute g and p_{at} into equation (A.2) to obtain the pressure of the water vapour p_s.

b. Substitute T_M into equation (A.1) to obtain the saturation vapour pressure p_{ss}.

c. Substitute p_s and p_{ss} into equation (A.5) to obtain the relative humidity φ.

d. Substitute g T_M, and P_{at} into equation (A.9) to obtain the specific volume v.

e. Substitute p_s into equation (A.10) to obtain the dew point temperature t_d.

f. Substitute g'_{ss} and T_M into equation (A.11), and solve the equation numerically for the wet bulb temperature t´.

A.3.6 Off-coil condition

The moisture content g, of the moist air at off-coil condition is equal to the moisture content of the supply air condition, because no water vapour is added or removed in the supply air duct. The dry bulb temperature T_C at the off-coil condition is equal to the assumed dry bulb temperature T_S of the supply air flow to the zone minus the temperature rise Δt_m of the electrical fan motor in the AHU and the rise in the dry bulb temperature Δt_d of the supply air due to the duct heat gain (i.e. 0.15° C). The temperature rise Δt_m of the electrical fan motor was calculated from,

$$\Delta t_m = \frac{\dot{Q}_m}{\dot{V}_a \rho_a c_p} \tag{A.25}$$

Having obtained the values of g and T_C at the off-coil condition, the other air properties at the off-coil condition were obtained as given in the following steps,

1. Substitute g and p_{at} into equation (A.2) to obtain the pressure of the water vapour p_s.

2. Substitute T_C into equation (A.1) to obtain the saturation vapour pressure p_{ss}.

3. Substitute p_s and p_{ss} into equation (A.5) to obtain the relative humidity φ.

4. Substitute g, T_C and p_{at} into equation (A.9) to obtain the specific volume v.

5. Substitute p_s into equation (A.10) to obtain the dew point temperature t_d.

6. Substitute g'_{ss} and T_C into equation (A.11), and solve the equation numerically for the wet bulb temperature t´.

7. Substitute T_O into equations (A.14) and (A.15) to obtain the specific enthalpy of dry air h_a and water vapour h_g respectively.

8. Substitute g, h_a and h_g into equation (A.13) to obtain the total specific enthalpy of the moist air h.

The summary of the results of the above calculations is shown in Tables A.1-A.6 below for zone 1.

A.4 Tabulated Results

For zone number 1	
Supply condition	
Dry-bulb temperature	14.50 degree C
Dew-point temperature	12.45 degree C
Supply flow rate	2.5249 cubic meter per second
Moisture content	0.008976 kg per kg of dry air
Dry specific enthalpy	14.58 kJ per kg of dry air
Moist specific enthalpy	22.69 kJ per kg of dry air
Total specific enthalpy	37.26 kJ per kg of dry air
Ambient condition	
Dry-bulb temperature	47.40 degree C
Wet-bulb temperature	27.09 degree C
Dew-point temperature	19.50 degree C
Ventilation flow rate	0.6450 cubic meter per second
Relative humidity	20.89 percent
Moisture content	0.014222 kg per kg of dry air
Specific volume	0.929033 cubic meter per kg of dry air
Dry specific enthalpy	47.71 kJ per kg of dry air
Moist specific enthalpy	36.81 kJ per kg of dry air
Total specific enthalpy	84.51 kJ per kg of dry air
Room condition	
Dry-bulb temperature	24.00 degree C
Wet-bulb temperature	17.07 degree C
Dew-point temperature	12.98 degree C
Supply flow rate	2.5249 cubic meter per second
Relative humidity	50.00 percent
Moisture content	0.009299 kg per kg of dry air

Specific volume	0.854550 cubic meter per kg of dry air
Dry specific enthalpy	24.14 kJ per kg of dry air
Moist specific enthalpy	23.36 kJ per kg of dry air
Total specific enthalpy	47.51 kJ per kg of dry air
Return condition	
Dry-bulb temperature	24.15 degree C
Moisture content	0.009299 kg per kg of dry air
Specific volume	0.854981 cubic meter per kg of dry air
Dry specific enthalpy	24.29 kJ per kg of dry air
Moist specific enthalpy	23.67 kJ per kg of dry air
Total specific enthalpy	47.96 kJ per kg of dry air
Mixture condition on-coil	
Dry-bulb temperature	29.61 degree C
Wet-bulb temperature	19.82 degree C
Dew-point temperature	14.74 degree C
Relative humidity	40.32 percent
Moisture content	0.010447 kg per kg of dry air
Specific volume	0.872264 cubic meter per kg of dry air
Dry specific enthalpy	29.79 kJ per kg of dry air
Moist specific enthalpy	26.70 kJ per kg of dry air
Total specific enthalpy	56.49 kJ per kg of dry air
Mixture condition off-coil	
Dry-bulb temperature	12.79 degree C
Wet-bulb temperature	12.57 degree C
Dew-point temperature	12.45 degree C
Relative humidity	97.60 percent
Moisture content	0.008976 kg per kg of dry air
Specific volume	0.821877 cubic meter per kg of dry air
Dry specific enthalpy	12.85 kJ per kg of dry air
Moist specific enthalpy	22.66 kJ per kg of dry air
Total specific enthalpy	35.51 kJ per kg of dry air
Zone load calculation	
Sensible heat gain	29.872 kW
Latent heat gain	2.428 kW
Sensible to total heat ratio	0.925
Fan motor heat gain	4.892 kW
Supply duct heat gain	0.469 kW
Return duct heat gain	0.455 kW
Fresh air heat gain	25.694 kW
Condensed water load	-0.237 kW
Cooling coil load	64.530 kW
Chilled water flow rate	2.768 L/s

Table A.1 Psychrometric properties of air at different states in zone 1.

A.5 References

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Appendix B

Equipment Specifications

Specifications	Ice Storage System		
	Partial		Full
	Load Levelling	50% Demand Limiting	
Model	TS120	TS240	TS240
Number of Ice Tank Required	1	1	2
Total Storage Capacity per Tank (kW_th)	120	240	240
Latent Capacity per Tank (kW_th)	109	210	210
Sensible Capacity per Tank (kW_th)	11	30	30
Maximum Pressure Drop (kPa)	117	117	117
Volume of Water per Tank (m^3)	4.50	8.33	8.33
Volume of Solution per Tank (m^3)	0.63	1.17	1.17
Diameter (cm)	226	256	256
Height (cm)	184	248	248
Weight, Empty (kg)	594	1032	1032
Weight, Total Operating (kg)	6190	10567	10567

Table B.1 Specification of ice storage tank (CALMAC)

Specifications	Ice Storage System		
	Partial		Full
	Load Levelling	50% Demand Limiting	
Model	TS120	TS240	TS240
Number of Ice Tank Required	1	1	2
Total Storage Capacity per Tank (kW_th)	120	240	240
Latent Capacity per Tank (kW_th)	109	210	210
Sensible Capacity per Tank (kW_th)	11	30	30
Maximum Pressure Drop (kPa)	117	117	117
Volume of Water per Tank (m^3)	4.50	8.33	8.33
Volume of Solution per Tank (m^3)	0.63	1.17	1.17
Diameter (cm)	226	256	256
Height (cm)	184	248	248
Weight, Empty (kg)	594	1032	1032
Weight, Total Operating (kg)	6190	10567	10567

Table B.2 Specification of ice storage tank (Dunham-Bush)

Chiller Specifications	Conventional	Ice Storage System			Chilled Water Storage System		
		Partial		Full	Partial		Full
		Load Levelling	50% Demand Limiting		Load Levelling	50% Demand Limiting	
Model	30GX152-PH3	30GX112-PH3	30GX132-PH3	30GX162-PH3	30GX082-PH3	30GX112-PH3	30GX122-PH3
Chiller Capacity (kW_t) Normal/Charging	422	306/205	354/246	429/286	226	311	334
Power Input (kW_e) Normal/Charging	206	142/126	187/164	219/190	110	143	166
COP Normal/Charging	2.04	2.1/1.6	1.84/1.41	1.9/1.4	2.06	2.17	2.01
Flow Rate (Ls^{-1}) Normal/Charging	16.5	12.3/12.7	14.4/15.3	17.4/17.8	5.41	7.42	7.99
No of compressors	2	2	2	2	2	2	2
Exit Temperature (Charging) (^oC)	-	-5	-5	-5	-	-	-
Inlet Temperature (Charging) (^oC)	-	-1	-1	-1	-	-	-
Exit Temperature (Normal) (^oC)	6.67	5.56	5.56	5.56	5.56	5.56	5.56
Inlet Temperature (Normal) (^oC)	12.78	11.67	11.67	11.67	15.56	15.56	15.56
Cooler Pressure drop (kPa) Normal/Charging	24	29/33	34/42	31/35	9	9	11
Working Fluid	Water	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	Water	Water	Water

Table B.3 Specifications of chillers

AC System			Coil Data												Fan		Motor	
	AHU No.	Total Pressure (Pa)	Flow Rate of the Supply Air (m^{3}h^{-1})	Air on-Coil Dry Bulb Temperature ($^{\circ}$C)	Air on-Coil Relative Humidity (%)	Air off-Coil Dry Bulb Temperature ($^{\circ}$C)	Air off-Coil Relative Humidity (%)	Sensible Capacity (kWt)	Total Capacity (kWt)	Maximum Capacity (kWt)	Volumetric Water Flow Rate (m^{3}h^{-1})	Inlet Water Temperature ($^{\circ}$C)	Outlet Water Temperature ($^{\circ}$C)	Water Pressure Drop (kPa)	Shaft Power (kW)	Efficiency (%)	Input Power (kW$_e$)	Efficienc y (%)
Conventional	1	839	9072	29.6	40	12.6	96	52.3	64.5	66.3	10.0	7.2	12.8	10	2.8	76.0	3.4	86.1
	2	829	9216	29.5	41	12.9	94	51.8	64.7	64.7	9.9	7.2	12.8	36	2.8	76.0	3.4	85.5
	3	921	10548	28.0	43	12.8	96	54.5	65.2	66.8	10.1	7.2	12.8	11	3.6	75.0	4.4	85.5
	4	908	10332	28.3	42	12.8	96	54.3	65.3	67.2	10.1	7.2	12.8	11	3.5	75.0	4.2	85.5
	5	875	9756	29.4	41	12.6	96	55.5	68.2	69.3	10.5	7.2	12.8	12	3.1	76.0	3.8	85.5
	6	899	10188	28.4	42	12.7	96	54.4	65.8	67.0	10.2	7.2	12.8	11	3.4	76.0	4.1	85.5
Ice Storage	1	843	9072	29.6	40	12.6	97	52.4	64.1	64.1	6.1	6.1	15.7	36	2.8	76.0	3.4	85.5
	2	850	9216	29.5	41	12.6	97	52.8	64.6	64.6	6.2	6.1	15.7	36	2.9	76.0	3.5	85.5
	3	925	10548	28.0	43	12.7	97	54.8	65.2	66.7	6.7	6.1	15.5	42	3.6	75.0	4.4	85.5
	4	910	10332	28.3	42	12.7	97	54.6	65.3	67.1	6.7	6.1	15.6	43	3.5	76.0	4.2	85.5
	5	891	9756	29.4	41	12.4	98	56.3	68.6	68.6	6.7	6.1	15.4	35	3.2	76.0	3.9	85.5
	6	899	10188	28.4	42	12.6	97	54.5	65.5	65.5	6.4	6.1	15.5	39	3.4	76.0	4.1	85.5
Chilled Water Storage	1	841	9072	29.6	40	12.5	97	52.5	64.5	64.5	5.8	7.2	16.8	19	2.8	76.0	3.4	86.1
	2	849	9216	29.5	41	12.6	97	52.8	64.5	64.5	5.7	7.2	16.9	18	2.9	76.0	3.5	85.5
	3	924	10548	28.0	43	12.7	97	54.8	65.3	65.3	6.7	7.2	16.9	26	3.6	75.0	4.4	85.5
	4	910	10332	28.0	42	12.4	97	54.6	65.3	65.9	5.9	7.2	16.7	42	3.5	76.0	4.2	85.5
	5	877	9756	29.4	41	12.6	98	55.7	68.2	68.8	6.2	7.2	16.7	30	3.1	76.0	3.8	85.5
	6	902	10188	28.4	42	12.6	97	54.6	65.7	65.7	5.8	7.2	16.9	27	3.4	76.0	4.1	85.5

Table B.4 Specifications of air handling units

Pumps Specifications	Conventional	Ice Storage System			Chilled Water Storage System		
		Partial		Full	Partial		Full
		Load Levelling	50% Demand Limiting		Load Levelling	50% Demand Limiting	
Model	NB 65-200/210	NB 80-315/302	NB 80-315/305	NB 65-315/309	NB 40-200/200	NB 40-250/245	NB 40-200/219
Number of Pump	2	2	2	2	2	2	2
Flow Rate Per Pump (Ls^{-1})	16.5	12.7	15.3	17.8	5.4	7.4	8.0
Head (m)	11.3	25.8	32.1	24.3	8.8	14.2	11.3
Pump Efficiency (%)	76.0	67.3	69.1	67.8	67.3	60.7	68.5
Motor Efficiency (%)	87.4	89.1	89.4	89.8	78.0	83.5	80.0
Motor Input Power (kW_e)	2.7	5.6	8.1	7.3	0.9	2.0	1.6
Working Fluid	Water	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	Water	Water	Water

Table B.5 Specifications of primary chilled water pumps

Pumps Specifications	Conventional	Ice Storage System			Chilled Water Storage System		
		Partial		Full	Partial		Full
		Load Levelling	50% Demand Limiting		Load Levelling	50% Demand Limiting	
Model	NB 40-250/245	NB 40-250/245	NB 40-250/245	NB 40-250/245	NB 32-200/219	NB 32-200/219	NB 32-200/219
Number of Pump	4	4	4	4	4	4	4
Flow Rate per Pump (Ls^{-1})	5.6	3.8	3.8	3.8	3.3	3.3	3.3
Head (m)	15.8	17.9	17.9	17.9	14.6	12.4	15.4
Pump Efficiency (%)	57.8	57.8	57.8	57.8	51.6	51.6	51.6
Motor Efficiency (%)	83.5	83.5	83.5	83.5	78	78	78
Motor Input Power (kW_e)	1.8	1.4	1.4	1.4	1.2	1.0	1.2
Working Fluid	Water	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	25% by mass Ethylene Glycol	Water	Water	Water

Table B.6 Specifications of secondary chilled water pumps

Appendix C

Chilled Water Storage Tank Design

C.1 Notation

Fr_i	Inlet Froude number	-
g	Acceleration of gravity	(ms^{-2})
h_i	Inlet opening height	(m)
L	Total length of the diffuser	(m)
q	Volume flow rate per unit diffuser length	(m^2s^{-1})
Q	Maximum volumetric flow rate	(m^3s^{-1})
Re_i	Inlet Reynolds number	-
ρ_i	Density of inlet water	(kgm^{-3})
ρ_w	Density of ambient water	(kgm^{-3})
υ	Kinematic viscosity of the inlet water	(m^2s^{-1})

C.2 Tank construction

The storage tanks in chilled water storage systems can be constructed in a cylindrical or rectangular shape. Because of the lower surface to volume ratio hence lower degree of thermal loss and lower construction cost per kW_th of the cylindrical tanks compared to the rectangular tanks for the same volume, the chilled water tanks in the chilled water storage systems were assumed having cylindrical shape and can be constructed from steel and located above the ground.

Dorgan (1994) argued that aboveground cylindrical steel tanks with typical height of 12 to 15 m are typically constructed with height to diameter ratio of 0.5 to 1.2. Since the height of the storage tank does not exceed the height of the building of about 8 m therefore a slightly higher values than 1.2 for the height to diameter ratio were chosen for the tanks when the chilled water systems are design for load levelling and 50% demand limiting operation strategies and height to diameter ratio of 1.2 for full storage operation strategy as given in Table C.1. Based on the selected height to diameter ratios for the storage tanks other dimensions of the tanks were calculated and are summarised in Table C.1.

Volume of a Cylinder (m^3)	76	102	146
Assumed height to Diameter Ratio	1.5	1.3	1.2
Tank Radius (m)	2.0	2.3	2.7
Tank Diameter (m)	4.0	4.6	5.4
Height (m)	6.0	6.0	6.4
Tank Surface (m^2)	101.1	121.7	154.0
Surface to Volume Ratio (m^{-1})	1.3	1.2	1.1

Table C.1 Detailed dimensions of the chilled water tanks

C.3 Diffusers design

The diffusers are the most important elements in the design of the chilled water storage tanks of the chiller water storage systems, so this section illustrates and discusses the design of the diffusers piping network for the chilled water tanks that were obtained for the load levelling and 50% demand limiting partial and full storage chilled water systems.

The basic dimensionless parameters that are commonly used for the design of the diffusers in a chilled water storage system are Reynolds and Froude numbers. Mixing on the inlet side of the thermocline depends on both the Froude and Reynolds numbers. The Froude number is defined as the dimensionless ratio of inertia force to the buoyancy force acting on a fluid, and the Reynolds number is defined as the dimensionless ratio of inertia force to the viscous force. For the diffusers, the inlet Reynolds number was defined in terms of the flow rate per unit length entering or exiting the diffuser and kinematic viscosity as follows:

$$\mathrm{Re}_i = \frac{q}{\upsilon} \tag{C.1}$$

According to (Dorgan, 1994), in tanks with a height greater than about 5m, Re_i can be limited from 400 to 850, and those with a height of 12 m or more Re_i up to 2000 can be accepted. Furthermore, (Dorgan, 1994) recommended that generally the upper limit of Re_i should not exceed 850 for all tank sizes. By selecting a reasonable value of 850 for Re_i and knowing the maximum temperature of the inletting water into the tank that corresponds to the minimum kinematic viscosity, Equation (C.1) was solved for volume flow rate per unit diffuser length q.

Flow rate per unit length is defined in terms of the total volume flow rate, and the diffuser length by

$$q = \frac{Q}{L} \tag{C.2}$$

Since the maximum volumetric flow rate Q through storage is known, Equation (C.2) was solved for the total diffuser length L. If a large value is selected for Re_i the diffuser is short, while the selection of a smaller value for Re_i results in a longer and more conservative diffuser (Dorgan, 1994). Moreover, the maximum volumetric flow rate Q in equation (C.2) is a function of the system design and can not be changed by diffuser design practice. However, the total diffuser length L is by definition a function of diffuser design. Therefore, the total length of the diffuser can be changed (e.g. increased or decreased) to change the volume flow rate per unit diffuser length q hence changing the Reynolds number Re_i and Froude number Fr_i as required.

Gravity currents, which are necessary for the proper performance of the tank, will form for Froude numbers Fr_i less than 1 with limited mixing, when Fr_i is above 2.0, the effects of mixing become apparent (Yoo, 1986). The Frounde number Fr_i criterion is used to determine the required inlet height of the diffuser h_i. The inlet opening height h_i as shown in Figure C.1, is defined as the distance occupied by the incoming flow when it leaves the diffuser and forms the gravity current. For the bottom diffusers, inlet opening height h_i is equal to the vertical distance between the tank floor and the top of the opening of the diffuser. For chilled water tank, the inlet Froude number Fr_i is defined as,

$$Fr_i = \frac{q}{\left[g h_i^3 \frac{\rho_i - \rho_w}{\rho_i} \right]^{0.5}} \tag{C.3}$$

A Froude number Fr_i of 0.7 was selected for design of the inlet opening height h_i. Assuming a single octagon diffuser system is selected for all tanks in the chilled water storage systems under study, the diffusers characteristics were obtained for each design operation strategy as shown in Table C.2.

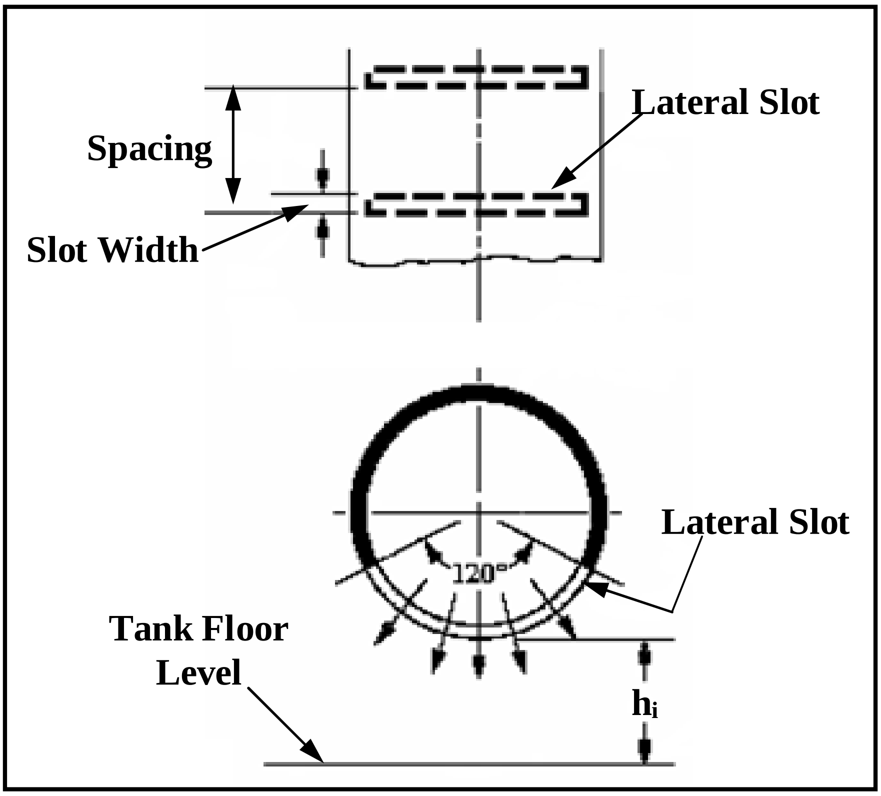

Figure C.1 Diffuser pipe cross section with 120° lateral slot.

Diffuser Description	Partial Storage		Full Storage
	Load Levelling	50% Demand Limiting	
Maximum Volumetric Flow Rate (Ls^{-1})	5.41	7.42	9.30
Assumed Reynolds Number Re_i	850	850	850
Inlet Water Kinematic Viscousity (m^2s^{-1})	1.50E-06	1.50E-06	1.45E-06
Volume Flow Rate Per Unit Diffuser Length (m^2s^{-1})	0.00127	0.00127	0.00123
Total Diffuser Length	4.3	5.8	7.6
Length of Each Side of the Octagon Diffuser (L)	0.5	0.7	0.9
Assumed Froude Number	0.7	0.7	0.7
Acceleration of Gravity (ms^{-2})	9.81	9.81	9.81
Density of Inlet Water (kgm^{-3})	999.8	999.8	999.8
Density of ambient Water (kgm^{-3})	998.7	998.7	998.7
Inlet Opening Height (cm)	6.7	6.7	6.6

Table C.2 Diffusers characterises for each design operating strategy in the chilled water storage system

In Table C.2, the volume flow rate per unit diffuser length q, was first computed based on assumed inlet Reynolds number Re_i of 850 using equation (C.1), and then the total diffuser length L was calculated using equation (C.2). The length of each section pipe of the octagon was calculated by dividing the total diffuser length L by 8. Based on Froude number Fr_i of 0.7, the inlet height h_i was obtained using equation (C.3). Furthermore, the density of the inlet water ρ_i was calculated at temperature of 5.56˚ C and the density of the ambient water ρ_w was evaluated at 16.67˚ C.

The characteristics of the pipe section in the octagon diffusers were further obtained based on maximum allowable velocity in the diffuser pipe. Table C.3 provides a description of one side pipe of the octagon diffusers in the chilled water thermal storage systems.

Pipe Description	Partial Storage		Full Storage
	Load Levelling	50% Demand Limiting	
Diffuser Pipe Nominal Diameter (mm)	80	100	100
Diffuser Pipe Internal Diameter (cm)	7.79	10.23	10.23
Diffuser Pipe External Diameter (cm)	8.89	11.43	11.43
Radial Length (120°) (cm)	8.16	10.71	10.71
Total Area of the Branch (cm^2)	47.7	82.1	82.1
Total Area of the Slots (cm^2)	21.7	31.6	37.3
Assumed Number of Slots	8	9	10
Flow Rate per Slot (Ls^{-1})	0.085	0.103	0.116
Slot Area (cm^2)	2.7	3.5	3.7
Slot Width (mm)	3.3	3.3	3.5
Maximum Flow Velocity at the Inlet to the Octagon (ms^{-1})	0.284	0.226	0.283
Maximum Flow Velocity in the Diffuser Pipe (ms^{-1})	0.142	0.113	0.142
Maximum Velocity Outlet Opening (ms^{-1})	0.312	0.294	0.311

Table C.3 Characteristics of single section pipe in the octagon diffuser

The nominal size of the diffuser pipe was obtained based on allowable maximum velocity in the diffusers of 0.274 ms^{-1} as recommended by (Fiorino, 1991). As shown in Table C.3, the calculated maximum velocity in the diffusers for all operation strategies was lower than 0.274 ms^{-1}. Furthermore, Fiorino (1991) argued that lowering the velocity in the diffuser precludes turbulent, jet-like flow near the diffusers openings.

Furthermore, the total slots area in the diffuser was calculated such that is equal or less than half of the cross sectional area of the inlet diffuser pipe to reduce the viscous pressure drop along the branch pipe, compared to the pressure drop across a representative opening. This tends to reduce the variation in flow rate through successive openings along a diffuser branch because the flow rate through an opening depends directly on the pressure drop across it (Wildin, 1990). The total slots area was modified to satisfy the conditions that the recommended maximum velocity at the slot opening is in the range of 0.3 to 0.6 ms^{-1} (Dorgan, 1994). Moreover, based on assumed number of slots, the slot cross sectional area for each slot was calculated and the slot width was obtained as shown in Table C.3 above.

C.4 References

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Appendix D

Chilled Water Pipe Design and Distribution System

D.1 Notation

c_{pw}	Specific heat capacity of chilled water	($Jkg^{-1}°K^{-1}$)
D	Inside diameter of pipe	(m)
f	Friction factor	-
g	Gravitational constant	(ms^{-2})
h	Head loss due to valve of fitting	(m)
h_f	Friction loss in pipe	(m)
K	Resistance coefficient	-
L	Length of pipe	(m)
$\dot{Q}_C$	Coil cooling capacity	(W)
Re	Reynolds number	-
T_{we}	Temperature of the exit chilled water from the cooling coil	(°C)
T_{wi}	Temperature of the inlet chilled water to the cooling coil	(°C)
v	Average velocity in pipe	(ms^{-1})
$\dot{V}_w$	Volumetric flow rate of water or glycol	(m^3s^{-1})
ε	Absolute roughness of pipe wall	(m)
µ	Absolute viscosity of the fluid	(Nsm^{-2})
ρ	Density of the fluid	(kgm^{-3})
ρ_w	Density of the fluid	(kgm^{-3})

D.2 Piping layout and arrangement

The chilled water distribution systems for conventional, ice storage, and chilled water storage were arranged with primary-secondary piping designs (James, 1996; ASHRAE, 2000a). This design is considered today the most popular chilled water system because it separates the chiller (that is, the chilled water production zone) from the distribution piping system (that is, the chilled water transportation zone) thereby reducing the differential pressure drop across the control valves of the cooling coil. However, this system has the disadvantage of requiring two sets of pumps, namely, primary and secondary pumps, and the further disadvantage that in the primary circuit some of the chilled water exiting the chiller returns to the chiller by the bypass circuit without using it, leading to a loss of energy in the primary pump.

The piping layout of the primary circuits for the conventional, ice storage, and chilled water storage in the plant room are shown in Figures D.1, D.2 and D.3 respectively. The designed volumetric flow rate in the primary circuit is met by a single centrifugal pump with a second standby pump of the same size. The primary pump is running 24 hours a day irrespective of the load on the chiller.

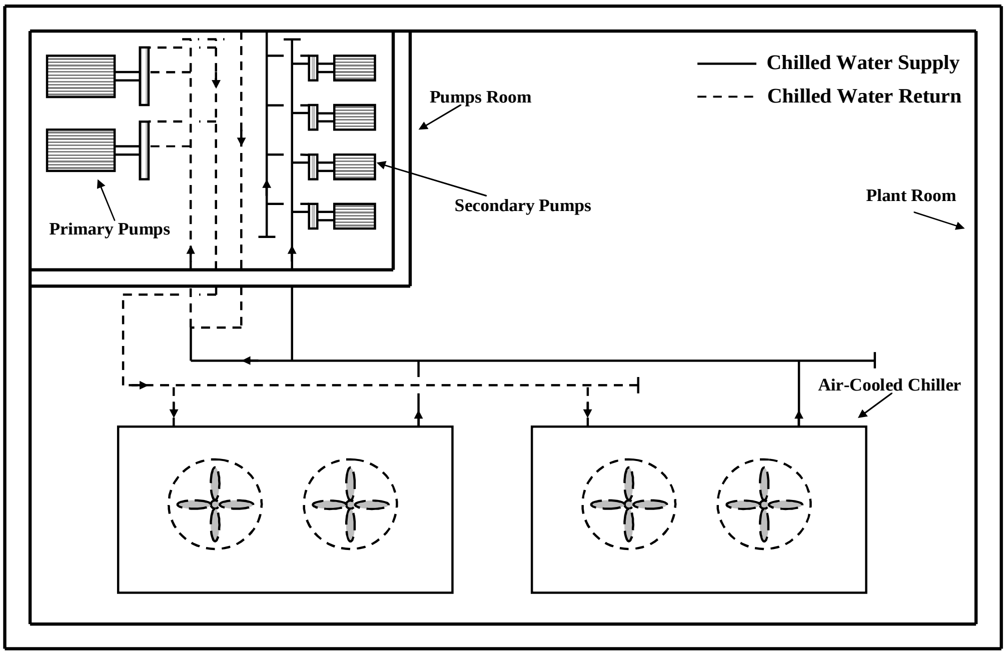

Figure D.1 Piping arrangement in the plant room for the conventional air conditioning system.

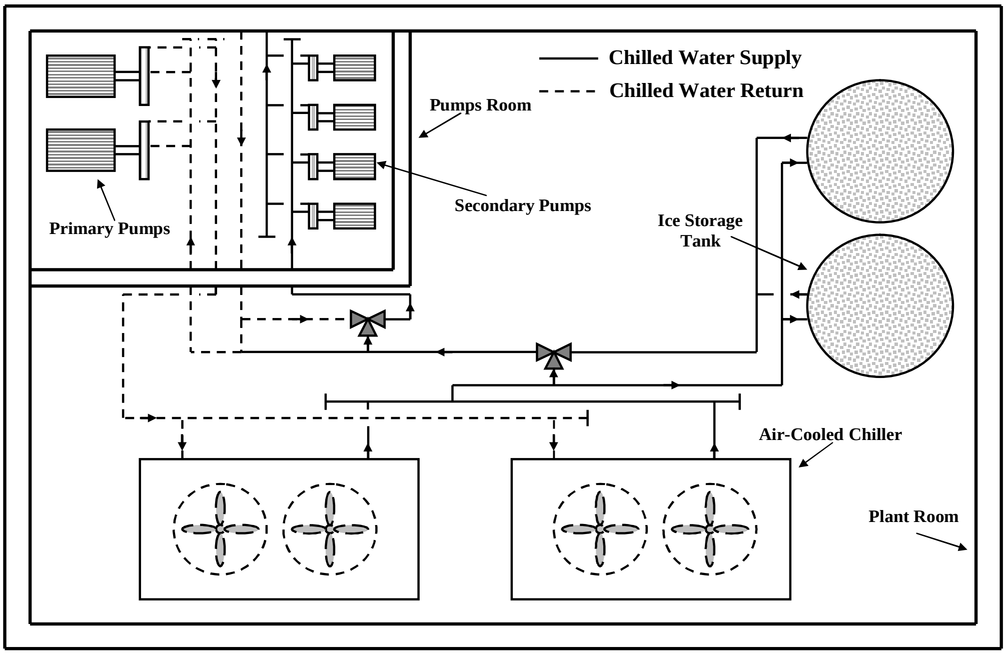

Figure D.2 Piping arrangement in the plant room for the ice thermal storage system.

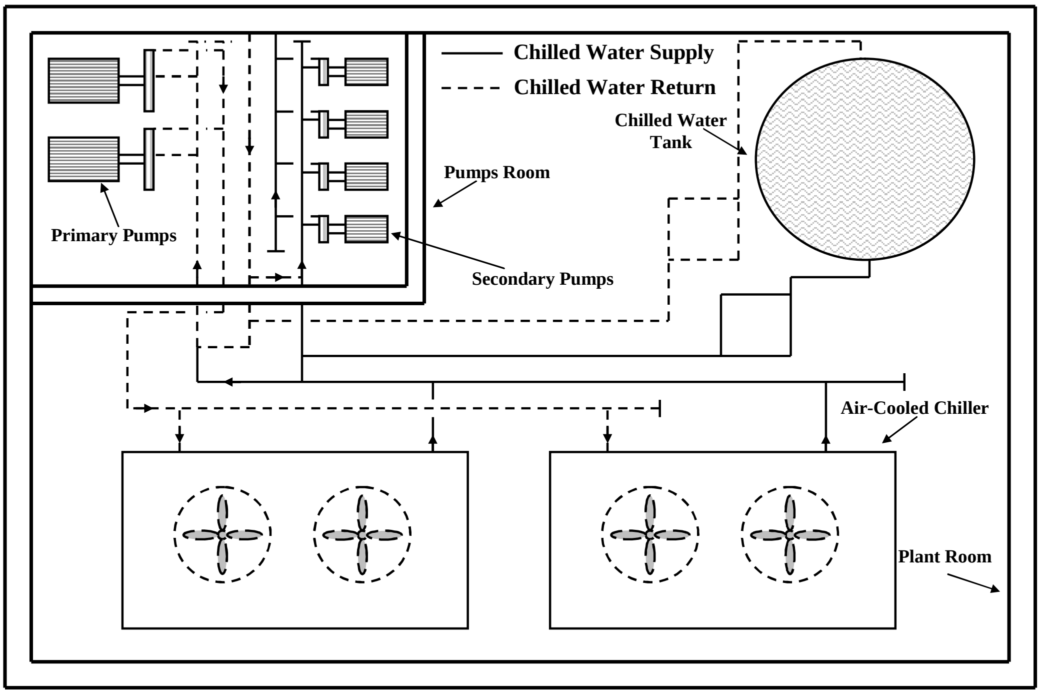

Figure D.3 Piping arrangement in the plant room of the chilled water thermal storage system.

In the ice thermal storage system, the primary pump must be capable of circulating the water solution through the ice storage tank during the discharging cycle. This is done by using the three-way valves as shown in Figure D.2. The function of the three-way valve is to divert the flow of the ethylene glycol solution through the ice tank to reduce further the temperature of the water solution to meet the building cooling load.

In the chilled water storage system, the primary pump is sized to overcome the pressure drop of the extra control valves that are associated with the chilled water tank and the pressure drop in the piping network of the diffusers within the tank. The primary pump delivers the required flow rate during both the charging and discharging cycles to the secondary circuit as shown in Figure D.3. When the chilled water storage system is at full storage, the primary pump is switched off completely, and the secondary pumps supply the water to the air handling units from the storage tank.

Figures D.1 to D.3 illustrate the number and locations of the secondary pumps in the plant room. The secondary pumps are located with the primary pumps in the pump room. There are four secondary pumps: three are operating and one is on standby. They are arranged in parallel where each pump takes its suction from a common header and discharges into another common header, thus sharing the flow while operating at the same head (Mackay, 2006). This pumping arrangement allows the pumps to be switched on and off as required to meet the varying demand. Each pump operates at the same head, but shares the flow rate with the other pumps. Multiple pumps in parallel can be controlled by either a flow measuring device or by a pressure differential controller (Levenhagen, 1992).

Figure D.4 shows the secondary piping arrangement on the roof of the clinic. This arrangement is same for the conventional, ice storage, and chilled water storage systems. The secondary chilled water distribution piping system is arranged with a direct return piping arrangement (ASHRAE, 2000b). This piping arrangement is strongly recommended by most designers for variable flow distribution and for where the flow rate through the cooling coil is controlled with a two-way temperature control valve (El-Amer, 2005). Furthermore, the direct return piping arrangement is the most economical arrangement for most modern buildings compared with other arrangements (for example, reverse return). It requires a minimum amount of piping and has less pipe friction than has an equivalent reverse return system (James, 1996).

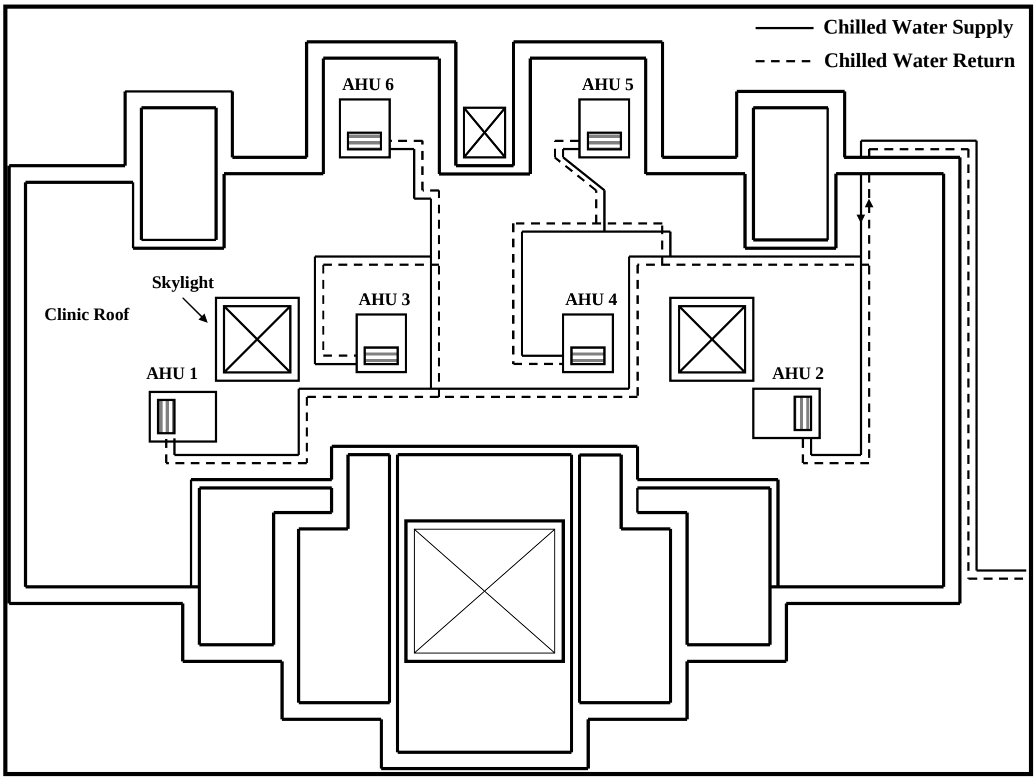

Figure D.4 Piping arrangement on the roof of the clinic.

D.3 Determination of the flow rate in the piping section

D.3.1 Flow rate in the secondary piping circuits

The volumetric flow rate in each section of the secondary chilled water pipes was obtained by calculating the volumetric flow of the chilled water flowing through the cooling coils using the following formula,

$$\dot{V}_w = \frac{\dot{Q}_C}{\rho_w c_{pw} (T_{wi} - T_{we})} \qquad \text{(D.1)}$$

For the conventional and chilled water thermal storage AC system, the density ρ_w and the specific heat c_{pw} of water was used in equation (D.1), and the density ρ_w and the specific heat c_{pw} for the ice thermal storage system were calculated for a 25% by mass of ethylene glycol obtained from their tabulated values in the ASHRAE handbook (ASHRAE, 2001b). Moreover, the volumetric flow rate of the chilled water $\dot{V}_w$ flowing through the cooling coils depends on the selected design of the temperature difference ($T_{wi} - T_{we}$) of the chilled water across the coils, and on the loads of the cooling coils. In the conventional AC system, the design temperature of the inlet chilled water T_{wi} to the cooling coil was taken about 0.56° C higher than the temperature of the chilled water exiting the chiller (MEW, 1999), and the exit temperature T_{we} was similar to that of the inlet chilled water temperature of the chiller. Based on the rating condition for Kuwait, the temperature of the chilled water exiting the chiller is 6.67° C and the temperature of the water entering the chiller is 12.78° C (MEW, 1999). Therefore, the design temperatures are 7.22° C for the inlet water T_{wi} and 12.78° C for the exit water T_{we}.

In ice storage AC systems, the inlet temperature T_{wi} of the chilled water to the cooling coil can be in the range of 1.1 to 7.22° C (Dorgan, 1994); low temperatures are usually selected for low temperature cold air distribution systems where the temperature of the air leaving the cooling coil is often between 5.6 to 8.3° C. Since most of the AC designers in Kuwait do not recommend designing the ducting system with low air temperatures because of the condensation that may result, it was assumed that the design of the ducting system was similar to that of conventional systems; hence, an inlet temperature T_{wi} of 6.12° C (that is, higher than the leaving chilled water temperature by 0.56°C) was selected. Since with the thermal storage system a temperature range across the cooling coil up to 13° C can be designed (Dorgan, 1994), the exit temperature T_{we} from the coil was arbitrarily selected and was assumed to be 15.01° C. The higher the exit temperature T_{we} from the cooling coils, the higher is ($T_{wi} - T_{we}$) and hence the lower volumetric flow rate $\dot{V}_w$ and power consumption of the secondary pumps.

In the chilled water storage system, the design chilled water temperature exiting the chiller during the charging cycle is 5.56° C, and by assuming that the temperature of the chilled water in the storage tank rises by about 1.1° C during the storing of the chilled water, and by adding 0.56° C to account for the heat gain in the piping system, the temperature of the inlet water T_{wi} to the cooling coil was calculated to be 7.22° C and the exit temperature it calculated to be 16.67° C, which is similar to the inlet temperature of the storage tank during the discharging cycle. The selection of higher exit temperature T_{we} from the cooling coil can take advantage of the lower pumping power. Table D.1 shows the maximum and design volumetric flow rates through each cooling coil for the conventional, ice storage, and chilled water storage systems, which are calculated based on the capacity of the cooling coils. Using the design of the volumetric flow rate, the flow rate throughout the piping network was calculated.

	Conventional System					
	Entering Coil Temperature (°C)	Exiting coil Temperature (°C)	Design Volumetric Flow Rate (Ls^{-1})	Maximum Volumetric Flow Rate (Ls^{-1})	Design coil Capacity (kW_t)	Maximum coil Capacity (kW_t)
AHU 1	7.22	12.78	2.77	2.77	64.5	64.7
AHU 2			2.78	2.78	64.7	64.8
AHU 3			2.80	2.81	65.2	65.6
AHU 4			2.80	2.82	65.3	65.6
AHU 5			2.93	2.93	68.2	68.2
AHU 6			2.82	2.83	65.8	66.0
	Ice Storage System					
	Entering Coil Temperature (°C)	Exiting coil Temperature (°C)	Design Volumetric Flow Rate (Ls^{-1})	Maximum Volumetric Flow Rate (Ls^{-1})	Design coil Capacity (kW_t)	Maximum coil Capacity (kW_t)
AHU 1	6.12	15.01	1.87	1.87	64.5	64.7
AHU 2			1.88	1.88	64.7	64.8
AHU 3			1.89	1.90	65.2	65.6
AHU 4			1.89	1.90	65.3	65.6
AHU 5			1.98	1.98	68.2	68.2
AHU 6			1.91	1.91	65.8	66.0
	Chilled Water Storage System					
	Entering Coil Temperature (°C)	Exiting coil Temperature (°C)	Design Volumetric Flow Rate (Ls^{-1})	Maximum Volumetric Flow Rate (Ls^{-1})	Design coil Capacity (kW_t)	Maximum coil Capacity (kW_t)
AHU 1	7.22	16.67	1.63	1.63	64.5	64.7
AHU 2			1.64	1.64	64.7	64.8
AHU 3			1.65	1.66	65.2	65.6
AHU 4			1.65	1.66	65.3	65.6
AHU 5			1.72	1.72	68.2	68.2
AHU 6			1.66	1.67	65.8	66.0

Table D.1 Calculated design and maximum volumetric flow rate and coil capacities in each AHU in the clinic

D.3.2 Flow rate in the primary circuit

The design volumetric flow rate in the primary circuit is equal to the chilled water flow rate through the chiller. This means that the volumetric flow rate in the primary circuit cannot be determined unless the refrigeration capacity, (or chiller capacity) is known and the design temperature difference across the chiller is specified. In order to determine the chiller size, the heat gain by the various components such as pipes, ducts, and the AHU and pump motors must be calculated and added to the building cooling load.

At this stage, the heat gain by the AHU fan motors and secondary chilled water pumps was determined; however, heat gain by the primary pumps and pipes could not be estimated because the flow rate was still unknown, therefore their values were first guessed as a percentage of the peak total system load (that is, building plus auxiliary loads), a reasonable estimate being 3% for the primary pumps motor and 4% for the piping system. In the case of ice and chilled water storage, higher values were estimated to account for the extra pressure drop in the storage tanks. Based on these assumptions, the chillers were sized and the volumetric flow rates were determined using ECAT2 Version 4.12 (Carrier, 2004).

D.4 Pipe sizing

The design volumetric flow rates having been determined in both the primary and the secondary circuits as discussed in section 5.4.2, the sizes of the pipes were then obtained based on the recommended volumetric flow rate range for a closed piping system as given in Table D.2 (El-Amer, 2005) for schedule 40 steel pipes.

The maximum and minimum volumetric flow rates given in Table D.2 are based on a maximum pressure drop of 391 Pam^{-1} and a maximum velocity of 3.1 ms^{-1}. Table D.2 also shows that the pressure drop for most of the water pipes is in the range of 98 Pam^{-1} to 391 Pam^{-1}, with a mean of 245 Pam^{-1}. Other guidelines for flow limitation in pipes available in studies by Carrier (1965), and by Crane (1988) can be used also to size the pipes properly. These are based on the type of service or the annual operating hours. Carrier (1965) recommends that the velocity does not exceed 4.6 ms^{-1} in any case. Stewart and Dona (1987) surveyed the literature relating to water flow rate limitations. Noise, erosion, and installation and operating costs all limit the maximum and minimum velocities in piping systems.

Pipe Size (mm)	Minimum Flow (Ls^{-1})	Maximum Flow (Ls^{-1})	Minimum Pressure Drop (Pam^{-1})	Maximum Pressure Drop (Pam^{-1})
15	0.00	0.13	0	391
20	0.19	0.25	245	391
25	0.32	0.47	196	391
32	0.50	1.01	122	391
40	1.07	1.51	196	391
50	1.58	3.03	122	391
65	3.09	4.86	196	391
80	4.92	8.83	147	391
100	8.90	17.67	122	391
125	17.73	31.55	147	391
150	31.61	50.48	171	391
200	50.54	107.27	98	391
250	107.33	157.75	122	269
300	157.81	227.16	122	220
350	227.22	265.02	122	196
400	265.08	347.05	98	171
450	347.11	441.70	88	147
500	441.76	567.90	78	122
600	567.96	820.30	59	98

Table D.2 Recommended volumetric water flow rate in a closed piping system

In the secondary piping circuit, the pipes that are directly connected to the AHUs were sized based on the maximum flow rate $\dot{V}_{wm}$ and other pipe sections in the secondary circuit were sized based on the design flow rate $\dot{V}_{wd}$. The maximum building cooling load including ventilation was determined based on the design day condition occurring at 1:00 pm, so the cooling coil capacities $\dot{Q}_C$ for each zone were calculated at the same hour. The coil capacity is equal to the summation of the heat gain by the zone, the supply and return ducts, the AHU motor and ventilation.

Equation (D.1) was then used to obtain the design flow rate $\dot{V}_{wd}$ for each cooling coil separately, and then the flow rates through each pipe sections from which the pipes were sized based on the recommended flow rates limitation as given in Table D.2.

D.5 The friction losses in pipes and fittings

It is essential to calculate the friction losses in the pipes and fittings in order to size properly the chilled water pumps. As water flows through pipes or valves, friction is generated that resists the flow. Energy is required to overcome this friction, and this energy is derived from the pumps. Therefore, calculating the friction losses in pipes and fittings will help to size the circulating chilled water pumps in the systems. Pressure losses in a piping system result from a number of system characteristics, which may be categorised as follows:

a. Pipe friction, which is a function of the surface roughness of the interior pipe, the inside diameter of the pipe, and the fluid velocity, density and viscosity.

b. Changes in the direction of the flow path.

c. Obstructions in the flow path.

d. Sudden or gradual changes in the cross-section and the shape of the flow path.

The pipe friction of the conventional, ice thermal storage and chilled water storage systems was calculated by using the Darcy-Weisbach formula i.e.

$$\mathrm{h}_f = f \frac{L}{D} \frac{v^2}{2g} \tag{D.2}$$

In equation (D.2), the friction factor f was calculated using the Colebrook equation

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log\left(\frac{2\varepsilon}{\mathrm{D}} + \frac{18.7}{\mathrm{Re}\sqrt{f}}\right) \tag{D.3}$$

Although there are other recently published equations that can be used specifically to obtain the friction factor f such as the Chen equation, the Churchill equation, the Haaland equation and the Swamee-Jain with a given Reynolds number Re and relative roughness ε/D (Janna, 1998), the Colebrook equation was preferred since historically it has been accepted as the most accurate functional representation of the Moody diagram in the turbulent zone (Hodge, 1985).

In equation (D.3), the pipe roughness ε, was 4.6×10^{-5} m, for the commercial steel pipe and the Reynolds number was a dimensionless number and was computed from the following equation,

$$\mathrm{Re} = \frac{\rho v D}{\mu} \qquad \text{(D.4)}$$

The fluid properties in equation (D.4) were evaluated at their operating temperatures. For the ice thermal storage design system, the Reynolds number Re, was calculated based on the density ρ, and the viscosity μ of a 25% by mass (for example, 22.9% by volume) ethylene glycol solution using ASHRAE coolants properties tables (ASHRAE, 2001a).

Most of the piping systems contain devices such as fittings and valves that cause losses sometimes called minor losses. Piping designed to carry chilled water in AC systems normally contains a relatively large number of fittings, valves, and other flow restricting devices. The losses due to these fittings may be quite high in comparison to the loss in straight pipes. Therefore, it is necessary to determine the type of fittings in the system and evaluate the losses associated with them in order to determine accurately the total system loss. The loss of pressure drop by a valve consists of (Crane, 1988):

a. The pressure drop within the valve itself.

b. The pressure drop in the upstream piping in excess of that which would normally occur if there were no valve in the line. This effect is small.

c. The pressure drop in the downstream piping in excess of that which would normally occur if there were no valve in the line. This effect may be comparatively large.

The valves in any piping system are the main fluid controlling elements; they can control the fluid flow either manually or automatically. The main functions of valves are (ASHRAE, 2000c)

a. Starting, stopping, and directing flow.
b. Regulating, controlling, or throttling flow.
c. Preventing backflow.
d. Relieving or regulating pressure.

The calculation of the pressure drop of the valves in the piping system was determined based on the number and types of valves that are associated with each component (for example, cooling coil, chiller, storage tank and so on) in the AC system, and within the piping system. The number and type of valves that must be installed with the different components in the AC system are shown in Appendix E (KEO, 2004) and those that control the flow within the piping systems are given in Appendix H. Classifications, types, construction, and functions of these types of valves are well explained in the ASHRAE handbook (ASHRAE, 2000c).

Extensive work has been done by Crane Company to develop comprehensive methods to evaluate the flow resistances of valves and fittings. The results of that company's work have been published (Crane, 1988). Most of the published results (Crane, 1988) were used to calculate the pressure drop of the valves and fittings in the piping system. The results found by Carrier (1965) were used also to calculate the pressure drop for some of the valves in the systems.

The friction loss for each valve and fitting in the piping systems was obtained by

$$h = K\frac{v^2}{2g} \qquad \text{(D.5)}$$

The resistance coefficient K was calculated for each valve type and size as well as fittings using the formulae available in Crane (Crane, 1988). The head loss in the equivalent number of metres of pipe for some of the valves was also obtained using tables and friction charts available in the piping design manual of Carrier (Carrier, 1965).

Head Loss	Conventional		Ice Storage System						Chilled Water Storage System					
			Partial				Full		Partial				Full	
			Load Levelling		50% Demand Limiting				Load Levelling		50% Demand Limiting			
	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.	Prim.	Secon.
Pipes (m)	2.1	8.0	2.7	4.4	1.4	4.4	2.6	4.4	2.5	4.2	3.8	3.6	2.6	4.0
Valves and Fittings (m)	6.7	6.8	13.7	10.0	7.9	10.0	13.6	10.0	5.4	6.2	9.4	4.5	7.6	7.1
Coil (m)	0.0	1.0	0.0	3.5	0.0	3.5	0.0	3.5	0.0	4.3	0.0	4.3	0.0	4.3
Chiller (m)	2.5	0.0	4.2	0.0	3.4	0.0	4.2	0.0	0.9	0.0	0.9	0.0	1.1	0.0
Storage Tank (m)	0.0	0.0	11.5	0.0	11.5	0.0	11.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Total (m)	11.3	15.8	32.1	17.9	24.3	17.9	32.0	17.9	8.8	14.6	14.2	12.4	11.3	15.4

Table D.3 Calculated pumping head in the primary and secondary circuits

The results of the friction losses in pipes, valves and fittings in the primary and secondary circuits of the conventional, ice storage, and chilled water storage systems are summarised in Table D.3.

Table D.3 also shows the head losses in the cooler of the chiller, cooling coil, and ice tank; the loss in the storage tanks (for example, diffusers pipes, and valves) of the chilled water storage systems were included with the losses in pipes and valves.

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Appendix E

Valves and Accessories

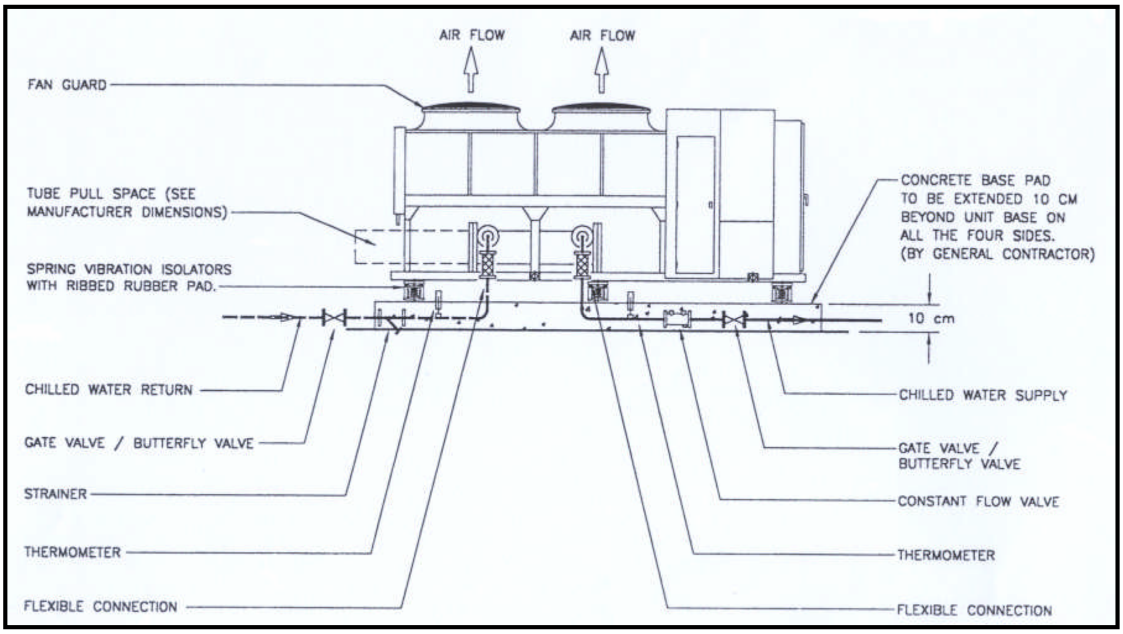

Figure E.1 Air-cooled chiller.

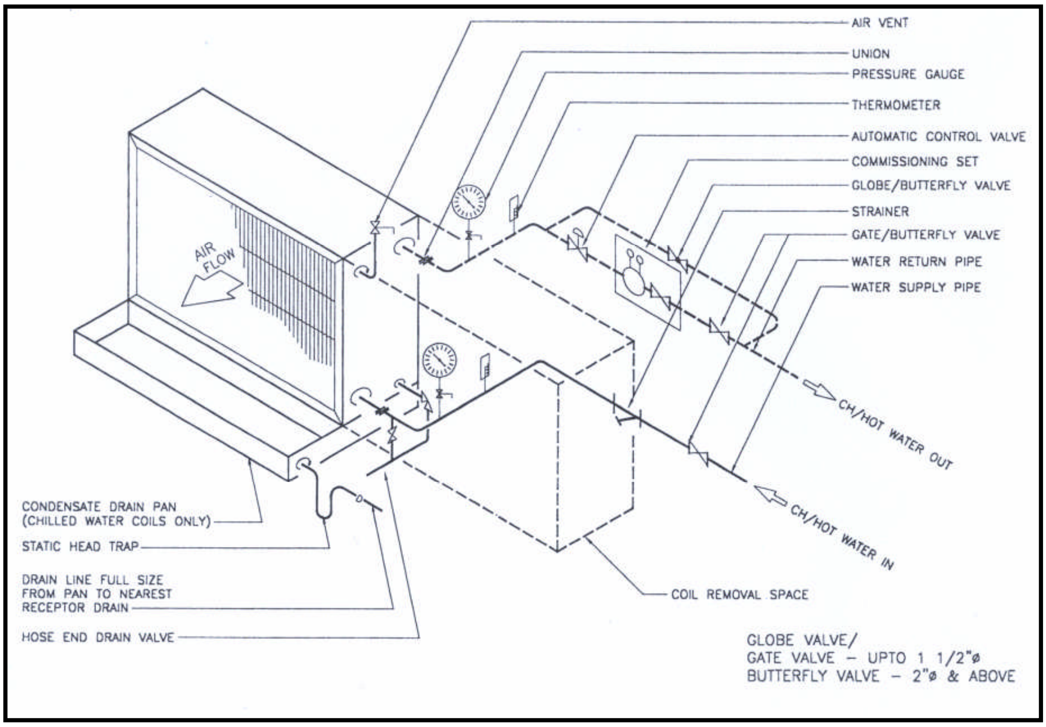

Figure E.2 Cooling coil.

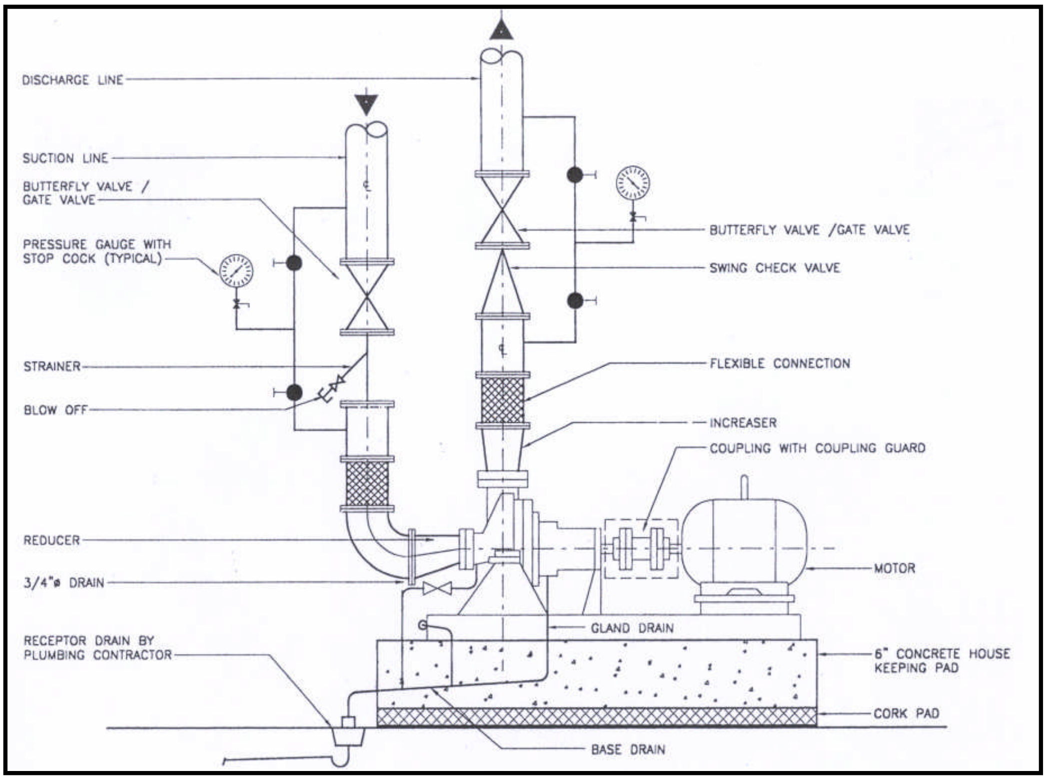

Figure E.3 Chiled water pump.

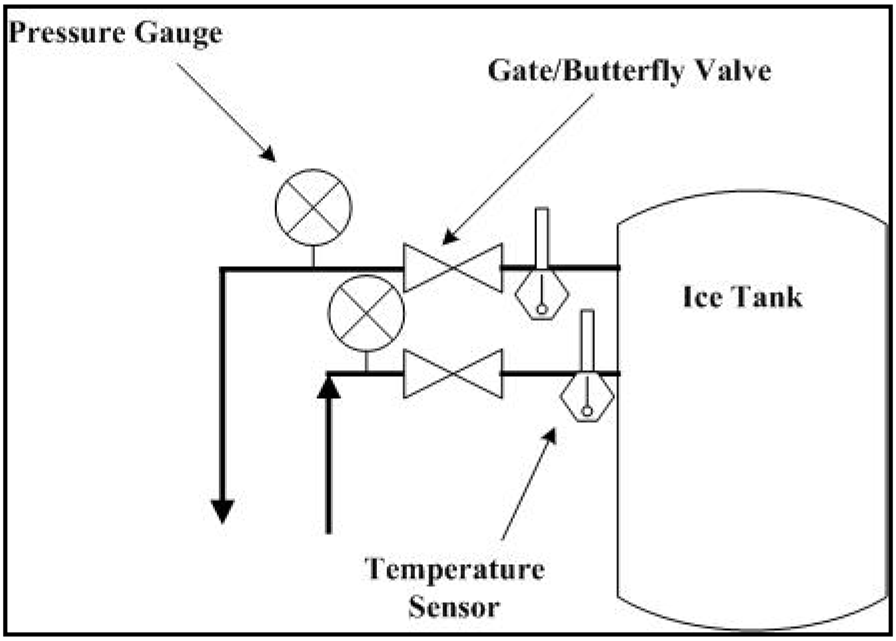

Figure E.4 Ice storage tank.

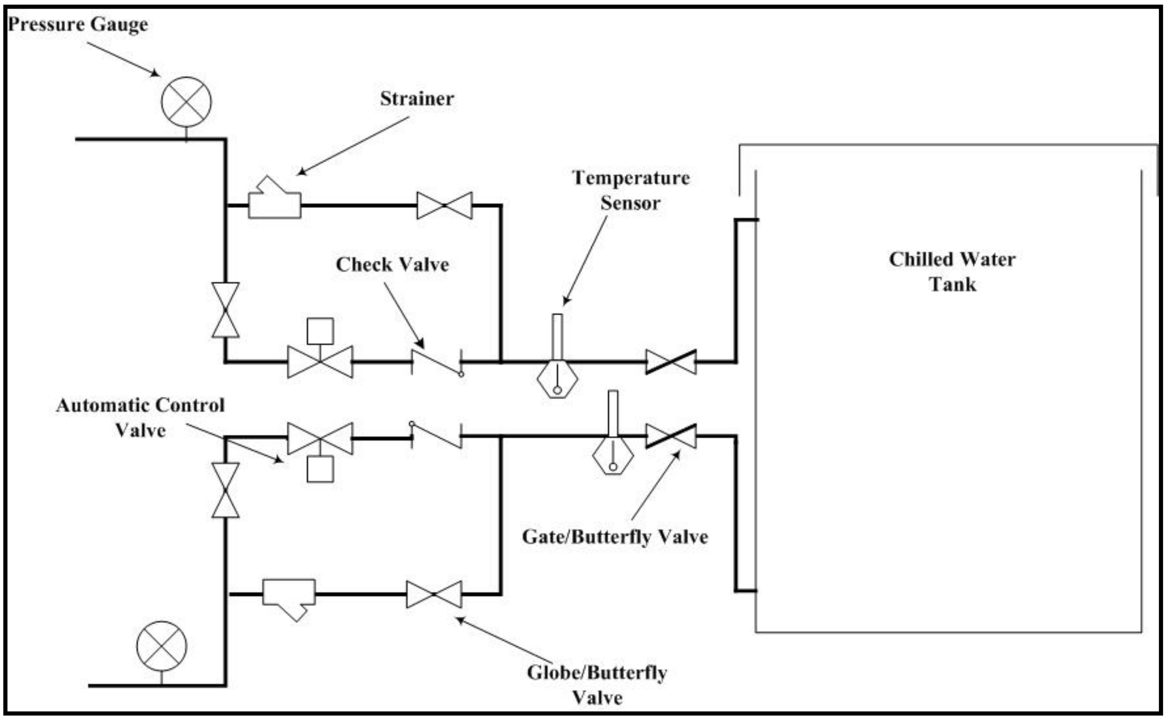

Figure E.5 Chilled water tank.

Appendix F

Chiller Performance

F.1 Conventional AC system

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)	20 (%)
47.4 (°C)	422	403	379	360	338	315	297	273	253	233	213	190	169	147	127	105	83
44.4 (°C)	443	423	398	378	355	331	312	286	265	244	223	199	177	155	133	111	89
41.4 (°C)	463	439	416	395	371	347	322	300	278	256	234	208	185	162	139	116	94
38.4 (°C)	483	458	434	409	388	363	337	314	290	267	242	217	193	169	145	121	99
35.4 (°C)	502	477	452	426	404	378	351	327	303	278	251	226	201	176	151	126	102
32.4 (°C)	521	496	470	442	416	393	365	340	315	289	261	235	208	182	157	131	109
29.4 (°C)	540	514	487	459	431	404	380	353	327	295	271	243	216	189	162	135	111
26.4 (°C)	559	531	504	475	447	419	394	360	333	305	280	251	223	195	168	140	115
23.4 (°C)	577	549	521	491	462	433	402	373	344	315	290	260	231	202	173	144	118
20.4 (°C)	595	566	537	507	477	448	415	385	356	326	299	267	238	208	178	149	123
17.4 (°C)	612	583	554	522	492	457	429	398	367	336	308	275	244	214	184	153	125

Table F.1 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)	20 (%)
47.4 (°C)	206	199	190	183	174	165	157	142	129	117	104	92	84	76	68	60	52
44.4 (°C)	199	192	183	176	167	158	150	136	124	112	100	88	80	73	65	58	51
41.4 (°C)	192	184	176	170	161	152	142	130	119	108	97	85	77	69	62	55	48
38.4 (°C)	185	178	170	162	155	146	136	125	114	104	92	82	74	66	59	53	47
35.4 (°C)	179	171	164	156	149	140	131	120	110	100	89	79	71	64	57	51	45
32.4 (°C)	173	166	158	150	142	135	125	116	106	96	85	76	68	61	54	48.5	43
29.4 (°C)	167	160	153	145	137	129	121	111	102	91	82	73	66	58	52	46.5	41.5
26.4 (°C)	162	155	148	140	132	124	116	105	97	88	80	71	63	56	49.8	44.6	39.6
23.4 (°C)	157	150	143	135	127	119	110	102	93	85	77	69	61	54	47.8	42.8	36.8
20.4 (°C)	153	146	139	131	123	115	106	98	90	82	75	66	59	52	45.9	41.2	36.5
17.4 (°C)	148	141	134	127	119	110	103	95	87	79	72	64	57	49.8	44.2	39.7	35.1

Table F.2 Absorbed power (kW_e)

F.2 Ice storage AC system (Load levelling)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	22.8 (%)
47.4 (°C)	306	290	275	259	144	230	215	199	184	169	153	138	122	107	92	68
44.4 (°C)	322	306	290	273	258	242	227	210	194	178	161	145	129	113	97	72
41.4 (°C)	339	322	305	288	271	255	237	222	205	187	169	153	135	119	102	76
38.4 (°C)	355	337	319	302	284	266	249	233	214	197	177	160	142	124	106	80
35.4 (°C)	370	352	334	315	298	278	259	242	224	204	185	167	148	130	111	84
32.4 (°C)	386	367	348	329	310	291	271	251	232	213	193	174	154	135	116	88
29.4 (°C)	401	381	362	342	321	300	282	262	242	222	200	180	160	140	120	92
26.4 (°C)	416	396	376	355	334	312	292	272	250	230	208	187	166	145	125	95
23.4 (°C)	430	410	389	365	346	324	303	280	259	239	215	194	172	151	129	99
20.4 (°C)	445	424	402	378	356	334	311	291	269	245	22	200	178	156	134	103
17.4 (°C)	459	437	415	391	368	345	322	301	278	254	229	207	184	160	137	107

Table F.3 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	22.8 (%)
47.4 (°C)	142	137	132	126	121	117	112	106	96	84	72	66	61	56	50	41.8
44.4 (°C)	137	132	127	121	116	112	107	101	91	81	69	64	58	53	48.1	39.9
41.4 (°C)	132	127	122	116	112	107	101	97	87	77	67	61	56	51	45.7	38.1
38.4 (°C)	127	122	117	112	107	102	97	92	83	74	64	59	54	48.5	43.5	36.5
35.4 (°C)	122	117	112	107	103	97	92	87	79	70	62	56	51	46.2	41.4	34.8
32.4 (°C)	118	113	108	103	98	93	88	83	75	67	59	54	49.1	44	39.4	33.3
29.4 (°C)	114	109	104	99	94	89	84	79	72	65	57	52	47	42	37.5	31.9
26.4 (°C)	110	105	100	95	90	85	80	75	68	62	55	50	45.1	40	35.7	30.5
23.4 (°C)	106	101	96	91	87	81	77	71	65	60	53	48.2	43.3	38.2	34.1	29.2
20.4 (°C)	102	98	93	88	83	78	73	68	63	57	52	46.4	41.5	36.5	32.6	28
17.4 (°C)	99	95	90	85	80	75	70	65	60	55	449.9	44.9	39.9	34.8	31	26.9

Table F.4 Absorbed power (kW_e)

F.3 Ice storage AC system (50% Demand limiting)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)	20 (%)
47.4 (°C)	354	336	320	302	284	266	249	231	214	196	177	159	142	124	106	89	70.8
44.4 (°C)	374	354	338	318	300	281	263	244	226	207	187	168	150	131	112	93	74.8
41.4 (°C)	393	372	356	335	315	296	276	257	237	218	196	177	157	138	118	98	78.6
38.4 (°C)	412	390	373	351	331	310	290	269	249	228	206	185	165	144	124	103	82.4
35.4 (°C)	431	408	387	367	346	325	301	282	260	239	215	194	172	151	129	108	86.2
32.4 (°C)	449	425	404	383	361	337	315	292	269	249	225	202	179	157	135	112	89.8
29.4 (°C)	467	442	421	399	376	351	328	304	280	257	234	210	187	163	140	115	93.4
26.4 (°C)	485	459	437	414	388	365	341	316	291	267	242	218	194	170	145	120	97
23.4 (°C)	502	476	453	427	403	378	354	328	302	276	251	226	201	176	150	126	100.4
20.4 (°C)	519	492	468	442	417	392	364	340	313	286	260	233	208	182	156	129	103.8
17.4 (°C)	536	509	484	457	431	403	377	349	323	295	268	241	214	188	161	134	107.2

Table F.5 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)	20
47.4 (°C)	187	179	172	164	156	148	140	129	118	107	94	86	78	70	63	56	49
44.4 (°C)	180	172	165	157	150	142	134	124	113	102	91	83	75	67	60	53	46
41.4 (°C)	173	165	159	151	143	136	128	118	108	98	87	79	72	64	58	51	45
38.4 (°C)	167	159	153	145	137	130	122	113	104	94	84	76	69	61	55	48.9	42.9
35.4 (°C)	161	153	146	139	132	124	116	108	99	91	81	73	66	58	53	46.8	41.8
32.4 (°C)	155	147	140	133	126	118	111	103	94	87	78	71	63	56	50	44.8	38.8
29.4 (°C)	149	142	135	128	121	113	106	98	91	83	75	68	61	53	48.1	42.9	38
26.4 (°C)	144	137	130	123	116	109	102	94	87	80	73	65	58	51	46	41	36
23.4 (°C)	139	132	125	118	111	104	97	91	84	77	70	63	56	48.7	44	39.3	34.6
20.4 (°C)	135	128	121	114	107	100	93	87	81	74	68	61	54	46.7	42.2	37.5	33
17.4 (°C)	131	123	117	110	103	96	89	83	78	72	66	59	52	44.7	40.5	35.9	31.7

Table F.6 Absorbed power (kW_e)

F.4 Ice storage AC system (Full)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	27.6 (%)
47.4 (°C)	429	409	388	364	345	322	300	281	256	237	215	193	172	150	129	117
44.4 (°C)	450	428	403	382	358	338	315	295	269	249	226	202	180	157	135	123
41.4 (°C)	470	448	421	400	375	354	330	306	281	260	236	212	188	165	141	129
38.4 (°C)	490	467	440	417	392	368	345	319	294	271	246	221	196	172	147	134
35.4 (°C)	510	486	458	435	408	384	359	333	306	282	256	230	204	179	153	140
32.4 (°C)	529	505	475	452	424	399	370	346	318	293	265	238	212	185	159	146
29.4 (°C)	548	523	493	468	440	412	385	359	330	304	275	247	219	192	164	151
26.4 (°C)	567	541	510	480	456	427	398	368	341	314	284	255	227	198	170	157
23.4 (°C)	585	558	527	496	469	439	412	381	353	325	293	263	234	205	176	162
20.4 (°C)	603	570	543	512	484	454	422	393	364	331	302	271	241	211	181	167
17.4 (°C)	620	587	560	528	499	468	435	405	375	341	311	279	248	217	186	0

Table F.7 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	27.6 (%)
47.4 (°C)	219	211	202	193	185	176	164	151	136	124	111	101	92	83	74	69
44.4 (°C)	211	203	193	185	176	169	157	146	131	120	107	97	89	80	71	66
41.4 (°C)	204	196	186	179	170	162	151	139	126	115	103	94	85	77	68	64
38.4 (°C)	197	189	180	172	163	155	145	133	122	111	99	90	82	74	65	61
35.4 (°C)	190	182	173	166	157	149	140	128	117	107	96	87	79	71	63	59
32.4 (°C)	184	176	167	160	151	144	133	124	113	103	93	84	76	68	60	57
29.4 (°C)	178	171	162	154	146	138	128	119	109	100	90	81	73	65	58	55
26.4 (°C)	172	165	156	148	141	133	124	114	105	97	87	79	70	63	56	53
23.4 (°C)	167	160	151	143	136	127	119	110	102	94	84	76	68	61	54	51
20.4 (°C)	163	154	147	139	131	123	114	107	99	90	82	74	66	58	52	49.4
17.4 (°C)	158	150	143	134	127	119	111	103	96	87	80	72	63	56	50	0

Table F.8 Absorbed power (kW_e)

F.5 Chilled water storage AC system (Load levelling)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	226	216	205	192	182	170	159	147	136	124	114	102	91	79	68	57
44.4 (°C)	239	228	216	204	191	180	167	155	144	131	121	108	96	84	72	60
41.4 (°C)	252	240	228	214	202	190	177	165	152	139	127	113	101	88	75	63
38.4 (°C)	264	250	237	225	212	197	184	171	160	147	132	119	106	92	79	66
35.4 (°C)	276	262	248	236	221	207	194	180	166	152	138	124	111	97	83	69
32.4 (°C)	288	274	260	245	231	217	203	188	174	160	145	130	115	101	87	72
29.4 (°C)	300	286	271	256	242	225	211	197	181	166	151	135	120	105	90	75
26.4 (°C)	312	297	282	267	250	235	219	204	189	173	157	140	125	109	94	78
23.4 (°C)	324	308	293	275	259	243	228	211	196	179	163	146	129	113	97	81
20.4 (°C)	335	319	301	286	269	251	235	219	202	185	169	151	134	117	100	84
17.4 (°C)	346	330	312	296	277	260	242	226	209	192	175	156	139	121	104	87

Table F.9 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	110	106	102	98	94	89	83	78	73	64	56	48	43.8	39.6	35.4	31.4
44.4 (°C)	106	102	98	94	89	84	79	74	69	61	54	46.2	42	37.8	33.6	29.6
41.4 (°C)	102	98	94	89	85	80	75	71	65	58	52	44.5	40.3	36.1	31.8	27.9
38.4 (°C)	98	94	89	86	81	76	71	66	62	56	48.8	42.9	38.7	34.4	30.2	26.4
35.4 (°C)	94	90	86	82	77	72	68	63	58	52	46.7	41.4	37.1	32.8	28.6	25
32.4 (°C)	91	87	82	78	74	69	65	59	55	49.9	44.8	39.9	35.6	31.3	27.1	23.6
29.4 (°C)	88	83	79	75	71	66	61	56	52	47.4	42.9	38.4	34.1	29.9	25.6	22.4
26.4 (°C)	84	80	76	72	67	63	58	53	49	45.3	41.2	37.1	32.8	28.5	24.3	21.2
23.4 (°C)	81	77	73	69	64	60	55	50	46.4	43	39.5	35.8	31.5	27.3	23	20.1
20.4 (°C)	79	75	70	66	62	57	52	47.7	44	40.9	37.9	34.6	30.3	26	21.7	19.1
17.4 (°C)	76	72	67	64	59	54	49.3	44.9	41.8	39.1	36.5	33.4	29.2	24.9	20.6	18.1

Table F.10 Absorbed power (kW_e)

F.6 Chilled water storage AC system (50% Demand limiting)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	311	295	279	263	247	233	218	203	186	170	156	140	124	109	93	78
44.4 (°C)	327	311	294	277	261	246	230	214	196	180	164	147	131	114	98	82
41.4 (°C)	344	327	309	292	275	259	242	226	207	189	173	155	138	121	103	86
38.4 (°C)	360	342	324	306	288	271	252	233	217	198	181	162	144	126	108	90
35.4 (°C)	376	357	339	320	302	282	264	245	228	207	189	169	151	132	113	94
32.4 (°C)	392	372	353	333	315	294	274	256	235	217	197	176	157	137	118	98
29.4 (°C)	407	387	367	347	328	307	286	265	245	225	205	183	163	142	122	102
26.4 (°C)	422	402	381	360	338	316	297	276	254	234	212	190	169	148	127	106
23.4 (°C)	437	416	395	374	351	328	309	285	262	241	220	197	175	153	131	109
20.4 (°C)	452	430	408	384	363	340	317	294	272	249	227	203	181	158	136	113
17.4 (°C)	466	444	421	396	373	349	326	305	281	258	234	210	186	163	140	116

Table F.11 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	143	138	133	127	122	117	112	107	96	85	74	67	62	56	51	45.2
44.4 (°C)	138	133	127	122	117	112	107	103	92	81	71	64	59	54	48.5	43.2
41.4 (°C)	133	128	122	117	112	107	103	98	88	78	68	62	57	51	46.2	41.1
38.4 (°C)	128	123	118	113	108	103	98	92	84	74	66	59	54	48.9	44	39.2
35.4 (°C)	123	118	113	108	103	98	93	88	80	71	63	57	52	46.7	41.9	37.3
32.4 (°C)	119	114	109	104	99	94	89	84	76	68	61	55	49.6	44.5	39.9	35.6
29.4 (°C)	115	110	105	100	95	90	85	79	72	66	59	53	47.6	42.5	37.9	33.9
26.4 (°C)	111	106	101	96	91	86	81	76	69	63	56	51	45.6	40.6	36.2	32.4
23.4 (°C)	107	102	97	93	87	82	78	72	66	60	54	48.9	43.8	38.7	34.5	30.9
20.4 (°C)	103	99	94	89	84	79	74	69	63	58	53	47.1	42	37	32.9	29.5
17.4 (°C)	100	95	91	85	80	75	70	66	61	56	51	45.5	40.4	35.4	31.4	28.2

Table F.12 Absorbed power (kW_e)

F.7 Chilled water storage AC system (Full)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	334	317	300	283	268	252	235	216	199	185	167	150	134	117	100	84
44.4 (°C)	353	335	317	298	283	266	248	228	210	195	177	159	141	124	106	88
41.4 (°C)	370	352	333	314	298	279	261	241	221	205	186	167	148	130	111	93
38.4 (°C)	388	369	349	329	312	293	273	253	232	215	194	175	155	136	117	97
35.4 (°C)	405	385	365	345	326	306	285	264	243	225	203	182	162	142	122	101
32.4 (°C)	422	402	381	360	339	318	297	276	253	234	212	190	169	148	127	106
29.4 (°C)	439	418	396	374	353	331	309	287	264	244	220	197	176	154	132	110
26.4 (°C)	456	433	411	389	366	343	321	295	274	249	229	205	182	159	137	114
23.4 (°C)	472	449	426	403	379	356	332	307	285	259	237	212	189	165	142	118
20.4 (°C)	488	464	441	416	392	367	342	318	295	268	245	219	195	171	146	122
17.4 (°C)	503	479	455	430	405	380	354	329	304	277	253	226	201	176	151	126

Table F.13 Capacity (kW_t)

	100 (%)	95 (%)	90 (%)	85 (%)	80 (%)	75 (%)	70 (%)	65 (%)	60 (%)	55 (%)	50 (%)	45 (%)	40 (%)	35 (%)	30 (%)	25 (%)
47.4 (°C)	166	161	155	149	143	136	128	118	105	94	82	70	65	59	54	47.4
44.4 (°C)	160	154	149	143	137	130	123	112	100	90	79	68	62	57	51	45.2
41.4 (°C)	154	149	143	137	132	124	117	107	96	87	75	65	60	54	48.5	43
38.4 (°C)	148	143	137	132	126	119	112	103	92	83	72	63	57	52	46.2	40.9
35.4 (°C)	143	138	132	127	121	114	107	98	88	80	70	61	55	49.4	44	39
32.4 (°C)	138	133	127	122	115	109	102	94	84	76	67	58	53	47.3	41.8	37.2
29.4 (°C)	133	128	122	117	111	104	97	89	81	73	64	56	51	45.2	39.8	35.5
26.4 (°C)	128	123	118	113	106	99	93	84	77	69	62	54	48.8	43.3	37.9	33.8
23.4 (°C)	124	119	114	108	102	95	88	81	74	66	60	52	47	41.5	36.1	32.3
20.4 (°C)	120	115	110	104	98	91	84	77	71	64	58	51	45.2	39.7	34.4	30.8
17.4 (°C)	116	111	106	101	94	88	81	74	68	61	56	49	43.6	38.1	32.9	29.5

Table F.14 Absorbed power (kW_e)

Appendix G

Capital Cost Breakdown Tables

Estimated costs were provided by suppliers and contractors throughout Kuwait in 2006.

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 422 kW_c	2	70,910	141,819	4,255	10,636	1,418	158,129
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 16.5 Ls^{-1} & head 11.3 m	2	1,141	2,283	68	171	23	2,545
2.2	Secondary pumps, capacity 5.6 Ls^{-1} & head 15.8 m	4	1,107	4,428	133	332	44	4,937
3	**Expansion tank, complete with all required fittings, controls , valves and accessories**							
3.1	Closed type 0.05 m^3	1	1,245	1,245	37	93	12	1,388
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 100 mm	1	761	761	23	57	8	848
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 66.3 kW_c	1	7,762	7,762	233	582	78	8,655
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.7 kW_c	1	7,797	7,797	234	585	78	8,693
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 66.8 kW_c	1	7,942	7,942	238	596	79	8,855
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 67.2 kW_c	1	7,942	7,942	238	596	79	8,855
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 69.3 kW_c	1	7,942	7,942	238	596	79	8,855
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 67.0kW_c	1	7,904	7,904	237	593	79	8,813
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, complete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	80 mm	75	19	1,426	43	107	14	1,589
6.3	100 mm	205	31	6,379	191	478	64	7,112
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm, 25 mm Thickness	109	6	658	20	49	7	733
7.2	80 mm, 50 mm Thickness	75	17	1,296	39	97	13	1,445
7.3	100 mm, 50 mm Thickness	205	20	4,111	123	308	41	4,584
8	**Aluminium cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	80 mm	14	21	297	9	22	3	331
8.3	100 mm	44	21	919	28	69	9	1,025
9	**Flexible pipe connections, complete with all required accessories**							
9.1	65 mm	8	31	249	7	19	2	278
9.2	100 mm	8	42	332	10	25	3	370
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	100 mm	1	398	398	12	30	4	444
11	**Commission set, complete with flow measuring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
11.2	100 mm	2	398	796	24	60	8	887
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	166	664	20	50	7	741
12.2	100 mm	2	176	353	11	26	4	393
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	93	374	11	28	4	417
13.3	100 mm	4	104	415	12	31	4	463

14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	8	31	249	7	19	2	278
14.3	100 mm	8	42	332	10	25	3	370
15	**Constant flow valves, complete with all accessories**							
15.1	100 mm	2	709	1,418	43	106	14	1,581
16	**Pressure gauge**							
16.1	50 mm	12	16	187	6	14	2	208
16.2	100 mm	4	16	62	2	5	1	69
17	**Pressure gauge with stop cock typical**							
17.1	80 mm	8	14	111	3	8	1	123
17.2	100 mm	4	14	55	2	4	1	62
18	**Thermometers**							
18.1	50 mm	12	48	581	17	44	6	648
18.2	100 mm	4	48	194	6	15	2	216
19	**Air Vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Ducting System**							
20.1	Ducts, labour, insulation, cladding, grills, diffusers, including all accessories							32,843
21	**Supply and installation of components**							
21.1	Control items, fittings, wiring, sensors and accessories including Building Automation System			15,050	0	1,129	151	16,329
22	**Electrical works associated with HVAC**							
22.1	Panels, switches, boards, cables, wiring, conducting and controls			6,226	187	467	62	6,942
23	**Mechanical identification systems**							
23.1	Valves, tags, labels, etc			173	5	13	2	193
24	**Testing , adjusting and commissioning**							
24.1	HVAC installations			692	21	52	7	771

Table G.1 Breakdown of the capital costs for conventional AC system

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 306 kW_c	2	59,322	118,644	3,559	8,898	1,186	132,288
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 12.7 Ls^{-1} & head 25.8 m	2	2,041	4,082	122	306	41	4,551
2.2	Secondary pumps, capacity 3.8Ls^{-1} & head 17.9m	4	1,107	4,428	133	332	44	4,937
3	**Expansion Tank, complete with all required fittings, controlls , valves and accessories**							
3.1	Closed type 0.37 m^3	1	1,730	1,730	52	130	17	1,928
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 100 mm	1	761	761	23	57	8	848
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.1 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.6 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 66.7 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 67.1 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.6 kW_c	1	8,046	8,046	241	603	80	8,971
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.5 kW_c	1	8,077	8,077	242	606	81	9,006
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, compelete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	65 mm	51	16	788	24	59	8	878
6.3	80 mm	24	19	463	14	35	5	516
6.4	100 mm	220	31	6,841	205	513	68	7,627

7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	658	20	49	7	733
7.2	65 mm	51	15	770	23	58	8	859
7.3	80 mm	24	17	421	13	32	4	469
7.4	100 mm	220	20	4,408	132	331	44	4,915
8	**Aluminum cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	65 mm	9	21	185	6	14	2	206
8.3	80 mm	5	21	95	3	7	1	106
8.4	100 mm	48	21	986	30	74	10	1,099
9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	100 mm	8	42	332	10	25	3	370
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	80 mm	1	346	346	10	26	3	386
11	**Commission set, complete with flow measring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	100 mm	2	176	353	11	26	4	393
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	100 mm	4	104	415	12	31	4	463
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	8	28	221	7	17	2	247
15	**3-way modulating valves**							
15.1	100 mm	2	1,038	2,075	62	156	21	2,314
16	**Pressure gauge**							
16.1	50 mm	12	16	187	6	14	2	208
16.2	100 mm	6	16	93	3	7	1	104
17	**Pressure gague with stop cock typical**							
17.1	65 mm	8	14	111	3	8	1	123
17.2	100 mm	4	14	55	2	4	1	62
18	**Thermometers**							
18.1	50 mm	12	48	581	17	44	6	648
18.2	100 mm	6	48	291	9	22	3	324
19	**Air Vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Ice storage tank**							
20.1	Storage capacity 272 kW_ch	1	17,641	17,641	529	1,323	176	19,670
21	**Glycol solution**	(kg)						
21.1	For 25% by mass ethylene glycol	933	2	1,840	55	138	18	2,051
22	**Supply and installation of components**							
22.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
23	**Supply and installation of components**							
23.1	Control items, fittings, wiring, sensors and accessories including Building Automation Systems			19,547	0	1,466	195	21,208
24	**Electrical works associated with HVAC**							
24.1	Panels, switches, boards, cables, wiring, conducting and controls			5,075	152	381	51	5,659
25	**Mechanical identification systems**							
25.1	Valves, tags, labels, etc.			173	5	13	2	193
26	**Testing , adjusting and commissioning**							
26.1	HVAC installations			692	21	52	7	771

Table G.2 Breakdown of the capital costs for ice storage AC system operating with load levelling strategy

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 344 kW_c	2	65,202	130,405	3,912	9,780	1,304	145,401
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 15.3 Ls-1 & head 32.1 m	2	2,594	5,189	156	389	52	5,785
2.2	Secondary pumps, capacity 3.8 Ls-1 & head 17.9 m	4	1,107	4,428	133	332	44	4,937
3	**Expansion Tank, complete with all required fittings, controls , valves and accessories**							
3.1	Closed type 0.37 m^3	1	1,730	1,730	52	130	17	1,928
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 100 mm	1	761	761	23	57	8	848
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.1 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.6 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 66.7 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 67.1 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.6 kW_c	1	8,046	8,046	241	603	80	8,971
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.5 kW_c	1	8,077	8,077	242	606	81	9,006
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, complete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	65 mm	51	16	788	24	59	8	878
6.3	80 mm	24	19	463	14	35	5	516
6.4	100 mm	220	31	6,841	205	513	68	7,627
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	658	20	49	7	733
7.2	65 mm	51	15	770	23	58	8	859
7.3	80 mm	24	17	421	13	32	4	469
7.4	100 mm	220	20	4,408	132	331	44	4,915
8	**Aluminium cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	65 mm	9	21	185	6	14	2	206
8.3	80 mm	5	21	95	3	7	1	106
8.4	100 mm	48	21	986	30	74	10	1,099
9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	100 mm	8	42	332	10	25	3	370
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	80 mm	1	346	346	10	26	3	386
11	**Commission set, complete with flow measuring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	100 mm	2	176	353	11	26	4	393
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	100 mm	4	104	415	12	31	4	463
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	8	28	221	7	17	2	247
15	**3-way modulating valves**							
15.1	100 mm	2	1,038	2,075	62	156	21	2,314
16	**Pressure gauge**							
16.1	50 mm	12	16	187	6	14	2	208
16.2	100 mm	6	16	93	3	7	1	104
17	**Pressure gauge with stop cock typical**							
17.1	65 mm	8	14	111	3	8	1	123
17.2	100 mm	4	14	55	2	4	1	62
18	**Thermometers**							
18.1	50 mm	12	48	581	17	44	6	648
18.2	100 mm	6	48	291	9	22	3	324

19	**Air Vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Ice storage tank**							
20.1	Storage capacity 715 kW_ch	1	22,484	22,484	675	1,686	225	25,069
21	**Glycol solution**	(kg)						
21.1	For 25% by mass ethylene glycol	933	2	1,840	55	138	18	2,051
22	**Supply and installation of components**							
22.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
23	**Supply and installation of components**							
23.1	Control items, fittings, wiring, sensors and accessories including Building Automation System			19,547	0	1,466	195	21,208
24	**Electrical works associated with HVAC**							
24.1	Panels, switches, boards, cables, wiring, conducting and controls			5,075	152	381	51	5,659
25	**Mechanical identification systems**							
25.1	Valves, tags, labels, etc.			173	5	13	2	193
26	**Testing , adjusting and commissioning**							
26.1	HVAC installations			692	21	52	7	771

Table G.3 Breakdown of the capital costs for ice storage AC system operating with 50% demand limiting strategy

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 416 kW_c	2	72,466	144,933	4,348	10,870	1,449	161,600
2	**Chilled water pumps**							
2.1	Primary pumps, Capacity 17.8 Ls^{-1} & head 24.3 m	2	2,387	4,773	143	358	48	5,322
2.2	Secondary pumps, capacity 3.8 Ls^{-1} & head 17.9 m	4	1,107	4,428	133	332	44	4,937
3	**Expansion Tank, complete with all required fittings, controls , valves and accessories**							
3.1	Closed type 0.42 m^3	1	1,730	1,730	52	130	17	1,928
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 125 mm	1	1,038	1,038	31	78	10	1,157
4.2	Secondary circuit, pipe size 125 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.1 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.6 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 66.7 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 67.1 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.6 kW_c	1	8,046	8,046	241	603	80	8,971
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.5 kW_c	1	8,077	8,077	242	606	81	9,006
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, complete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	65 mm	51	16	788	24	59	8	878
6.3	80 mm	33	19	622	19	47	6	694
6.4	100 mm	136	31	4,239	127	318	42	4,727
6.5	125 mm	84	42	3,492	105	262	35	3,894
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	658	20	49	7	733
7.2	65 mm	51	15	770	23	58	8	859
7.3	80 mm	33	17	565	17	42	6	630
7.4	100 mm	136	20	2,732	82	205	27	3,046
7.5	125 mm	84	23	1,964	59	147	20	2,190
8	**Aluminium cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	65 mm	9	21	185	6	14	2	206
8.3	80 mm	6	21	131	4	10	1	146
8.4	100 mm	29	21	610	18	46	6	680
8.5	125 mm	20	21	423	13	32	4	472

9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	125 mm	8	52	415	12	31	4	463
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	80 mm	1	346	346	10	26	3	386
11	**Commission set, complete with flow measuring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	125 mm	2	176	353	11	26	4	393
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	125 mm	4	104	415	12	31	4	463
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	8	28	221	7	17	2	247
14.3	125 mm	8	52	415	12	31	4	463
15	**3-way modulating valves**							
15.1	100 mm	1	1,038	1,038	31	78	10	1,157
15.2	125 mm	1	1,176	1,176	35	88	12	1,311
16	**Pressure gauge**							
16.1	50 mm	12	16	187	6	14	2	208
16.2	125 mm	6	16	93	3	7	1	104
17	**Pressure gauge with stop cock typical**							
17.1	65 mm	8	14	111	3	8	1	123
17.2	125 mm	4	14	55	2	4	1	62
18	**Thermometers**							
18.1	50 mm	12	48	581	17	44	6	648
18.2	125 mm	6	48	291	9	22	3	324
19	**Air Vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Ice storage tank**							
20.1	Storage capacity 1529 kWch	2	21,273	42,546	369	123	123	43,161
21	**Glycol solution**	(kg)						
21.1	For 25% by mass ethylene glycol	1165	2	2,297	69	172	23	2,561
22	**Supply and installation of components**							
22.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
23	**Supply and installation of components**							
23.1	Control items, fittings, wiring, sensors and accessories including Building Automation System			19,547	0	1,466	195	21,208
24	**Electrical works associated with HVAC**							
24.1	Panels, switches, boards, cables, wiring, conducting and controls			6,137	184	460	61	6,843
25	**Mechanical identification systems**							
25.1	Valves, tags, labels, etc.			173	5	13	2	193
26	**Testing , adjusting and commissioning**							
26.1	HVAC installations			692	21	52	7	771

Table G.4 Breakdown of the capital costs for ice storage AC system operating with full strategy

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 226 kW_c	2	56,209	112,418	3,373	8,431	1,124	125,346
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 5.4 Ls^{-1} & head 8.8 m	2	899	1,799	54	135	18	2,006
2.2	Secondary pumps, capacity 3.3 Ls^{-1} & head 14.6 m	4	847	3,390	102	254	34	3,780
3	**Expansion Tank, complete with all required fittings, controls , valves and accessories**							
3.1	Closed type 1.05 m^3	1	4,151	4,151	125	311	42	4,628
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 80 mm	1	726	726	22	54	7	810
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.5 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.5 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 65.3 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 65.9 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.8 kW_c	1	8,077	8,077	242	606	81	9,006
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.7 kW_c	1	8,077	8,077	242	606	81	9,006
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, complete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	65 mm	51	16	788	24	59	8	878
6.3	80 mm	165	19	3,136	94	235	31	3,497
6.4	100 mm	102	31	3,166	95	237	32	3,530
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	677	20	51	7	754
7.2	65 mm	51	15	770	23	58	8	859
7.3	80 mm	144	17	2,483	74	186	25	2,768
7.4	100 mm	102	20	2,040	61	153	20	2,275
8	**Aluminium cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	65 mm	9	21	185	6	14	2	206
8.3	80 mm	27	21	569	17	43	6	634
8.4	100 mm	22	21	457	14	34	5	509
9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	80 mm	8	31	249	7	19	2	278
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	80 mm	8	346	2,767	83	208	28	3,085
11	**Commission set, complete with flow measuring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	80 mm	4	166	664	20	50	7	741
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	80 mm	6	93	560	17	42	6	625
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	4	28	111	3	8	1	123
14.3	80 mm	10	31	311	9	23	3	347
15	**Pressure gauge**							
15.1	50 mm	12	16	187	6	14	2	208
15.2	80 mm	2	16	31	1	2	0	35
16	**Pressure gauge with stop cock typical**							
16.1	65 mm	8	14	111	3	8	1	123
16.2	80 mm	4	14	55	2	4	1	62
17	**Thermometers**							
17.1	50 mm	12	48	581	17	44	6	648
17.2	80 mm	6	48	291	9	22	3	324
18	**Pressurised chilled water tank**							
18.1	Storage volume 76 m^3	1	27,250	27,250	817	2,044	272	30,783

19	**Air vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Supply and installation of components**							
20.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
21	**Supply and installation of components**							
21.1	Control items, fittings, wiring, sensors and accessories including BAS			26,915	0	2,019	269	29,202
22	**Electrical works associated with HVAC**							
22.1	Panels, switches, boards, cables, wiring, conducting and controls			3,334	100	250	33	3,718
23	**Mechanical identification systems**							
23.1	Valves, tags, labels, etc.			173	5	13	2	193
24	**Testing , adjusting and commissioning**							
24.1	HVAC installations			865	26	65	9	964

Table G.5 Breakdown of the capital costs for chilled water storage AC system operating with load levelling strategy

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 311 kW_c	2	59,668	119,336	3,580	8,950	1,193	133,059
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 7.4 Ls^{-1} & head 14.2 m	2	1,107	2,214	66	166	22	2,468
2.2	Secondary pumps, capacity 3.3 Ls^{-1} & head 12.4 m	4	847	3,390	102	254	34	3,780
3	**Expansion tank, complete with all required fittings, controlls , valves and accessories**							
3.1	Closed type 1.4 m^3	1	4,843	4,843	145	363	48	5,400
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 80 mm	1	726	726	22	54	7	810
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.5 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.5 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 65.3 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 65.9 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.8 kW_c	1	8,077	8,077	242	606	81	9,006
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.7 kW_c	1	8,077	8,077	242	606	81	9,006
6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, compelete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	13	1,466	44	110	15	1,635
6.2	65 mm	51	15	770	23	58	8	859
6.3	80 mm	144	17	2,490	75	187	25	2,776
6.4	100 mm	129	20	2,584	78	194	26	2,881
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	658	20	49	7	733
7.2	65 mm	51	15	776	23	58	8	865
7.3	80 mm	144	17	2,490	75	187	25	2,777
7.4	100 mm	102	20	2,046	61	153	20	2,282
8	**Aluminum cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	19	3	282
8.2	65 mm	9	21	185	6	14	2	206
8.3	80 mm	27	21	569	17	43	6	634
8.4	100 mm	22	21	457	14	34	5	509
9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	80 mm	8	31	249	7	19	2	278

10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	80 mm	8	346	2,767	83	208	28	3,085
11	**Commission set, complete with flow measring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	80 mm	4	166	664	20	50	7	741
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	80 mm	6	93	560	17	42	6	625
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	4	28	111	3	8	1	123
14.3	80 mm	10	31	311	9	23	3	347
15	**Pressure gauge**							
15.1	50 mm	12	16	187	6	14	2	208
15.2	80 mm	2	16	31	1	2	0	35
16	**Pressure gague with stop cock typical**							
16.1	65 mm	8	14	111	3	8	1	123
16.2	60 mm	4	14	55	2	4	1	62
17	**Thermometers**							
17.1	50 mm	12	48	581	17	44	6	648
17.2	80 mm	6	48	291	9	22	3	324
18	**Pressuriesed chilled water tank**							
18.1	Storage volume 102 m^3	1	33,717	33,717	1,012	2,529	337	37,995
19	**Air vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Supply and installation of components**							
20.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
21	**Supply and installation of components**							
21.1	Control items, fittings, wiring, sensors and accessories including Building Autmation System			27,779	0	2,083	278	30,141
22	**Electrical works associated with HVAC**							
22.1	Panels, switches, boards, cables, wiring, conducting and controls			4,588	138	344	46	5,116
23	**Mechanical identification systems**							
23.1	Valves, tags, labels, etc.			173	5	13	2	193
24	**Testing , adjusting and commissioning**							
24.1	HVAC installations			865	26	65	9	964

Table G.6 Breakdown of the capital costs for chilled water storage AC system operating with 50% demand limiting strategy

Item No.	Description	Qty	Unit	Unit Total	Labour	Over Head	Profit	Total
1	**Water chillers**							
1.1	Air cooled water chiller, unit size 334 kW_c	2	63,992	127,983	3,840	9,599	1,280	142,701
2	**Chilled water pumps**							
2.1	Primary pumps, capacity 8.0 Ls^{-1} & head 11.3 m	2	917	1,833	55	137	18	2,044
2.2	Secondary pumps, capacity 3.3 Ls^{-1} & head 15.4 m	4	847	3,390	102	254	34	3,780
3	**Expansion tank, complete with all required fittings, controls , valves and accessories**							
3.1	Closed type 1.99 m^3	1	4,843	4,843	145	363	48	5,400
4	**Air separator complete with all required fittings, valves, and accessories**							
4.1	Primary circuit, pipe size 80 mm	1	726	726	22	54	7	810
4.2	Secondary circuit, pipe size 100 mm	1	761	761	23	57	8	848
5	**Air handling units, with necessary sections including fan, cooling coil, mixing box, filters, motor, and all required accessories**							
5.1	AHU-01 - Air flow rate 9072 m^3h^{-1}& maximum capacity 64.5 kW_c	1	7,897	7,897	237	592	79	8,805
5.2	AHU-02 - Air flow rate 9216 m^3h^{-1}& maximum capacity 64.5 kW_c	1	8,077	8,077	242	606	81	9,006
5.3	AHU-03 - Air flow rate 10548 m^3h^{-1}& maximum capacity 65.3 kW_c	1	8,077	8,077	242	606	81	9,006
5.4	AHU-04 - Air flow rate 10332 m^3h^{-1}& maximum capacity 65.9 kW_c	1	8,077	8,077	242	606	81	9,006
5.5	AHU-05 - Air flow rate 9756 m^3h^{-1}& maximum capacity 68.8 kW_c	1	8,077	8,077	242	606	81	9,006
5.6	AHU-06 - Air flow rate 10188 m^3h^{-1}& maximum capacity 65.7 kW_c	1	8,077	8,077	242	606	81	9,006

6	**Chilled water piping of seamless black steel heavy duty schedule 40 pipes, complete with all required fittings, hangers, supports, sleeves, and all other accessories**	(m)						
6.1	50 mm	109	10	1,128	34	85	11	1,257
6.2	65 mm	51	16	788	24	59	8	878
6.3	80 mm	118	19	2,245	67	168	22	2,504
6.4	100 mm	162	31	5,059	152	379	51	5,640
7	**Chilled water pipes thermal insulation, complete with all required accessories**	(m)						
7.1	50 mm	109	6	658	20	49	7	733
7.2	65 mm	51	15	770	23	58	8	859
7.3	80 mm	118	17	2,041	61	153	20	2,276
7.4	100 mm	128	20	2,572	77	193	26	2,868
8	**Aluminium cladding for outdoor pipes**	(m^2)						
8.1	50 mm	12	21	253	8	1	0	261
8.2	65 mm	9	21	185	6	0	0	191
8.3	80 mm	23	21	467	14	1	0	482
8.4	100 mm	28	21	573	17	1	0	591
9	**Flexible pipe connections, complete with all required accessories**	(m)						
9.1	65 mm	8	28	221	7	17	2	247
9.2	80 mm	8	31	249	7	19	2	278
10	**2- way modulating valves, complete with actuators and all required accessories**							
10.1	50 mm	6	294	1,764	53	132	18	1,967
10.2	100 mm	8	398	3,182	95	239	32	3,548
11	**Commission set, complete with flow measuring taps and all required accessories**							
11.1	50 mm	6	242	1,453	44	109	15	1,620
12	**Swing check valves, complete with all accessories**							
12.1	65 mm	4	97	387	12	29	4	432
12.2	80 mm	2	166	332	10	25	3	370
12.3	100 mm	2	176	353	11	26	4	393
13	**Y- type strainer, complete with all accessories**							
13.1	50 mm	6	86	519	16	39	5	579
13.2	65 mm	4	90	360	11	27	4	401
13.3	80 mm	4	93	374	11	28	4	417
13.4	100 mm	2	104	208	6	16	2	231
14	**Butterfly valves**							
14.1	50 mm	18	21	374	11	28	4	417
14.2	65 mm	4	28	111	3	8	1	123
14.3	80 mm	8	31	249	7	19	2	278
14.4	100 mm	2	42	83	2	6	1	93
15	**Pressure gauge**							
15.1	50 mm	12	16	187	6	14	2	208
15.2	100 mm	2	16	31	1	2	0	35
16	**Pressure gauge with stop cock typical**							
16.1	65 mm	8	14	111	3	8	1	123
16.2	80 mm	4	14	55	2	4	1	62
17	**Thermometers**							
17.1	50 mm	12	48	581	17	44	6	648
17.2	80 mm	4	48	194	6	15	2	216
17.3	100 mm	2	48	97	3	7	1	108
18	**Pressurised chilled water tank**							
18.1	Storage volume 146 m^3	1	46,052	46,052	1,382	3,454	461	51,748
19	**Air vent**							
19.1	50 mm	6	21	125	4	9	1	139
20	**Supply and installation of components**							
20.1	Control items, fittings, wiring, sensors and accessories including Building Automation System							32,843
21	**Supply and installation of components**							
21.1	Control items, fittings, wiring, sensors and accessories including Building Automation System			27,018	0	2,026	270	29,315
22	**Electrical works associated with HVAC**							
22.1	Panels, switches, boards, cables, wiring, conducting and controls			4,927	148	370	49	5,494
23	**Mechanical identification systems**							
23.1	Valves, tags, labels, etc.			173	5	13	2	193
24	**Testing , adjusting and commissioning**							
24.1	HVAC installations			865	26	65	9	964

Table G.7 Breakdown of the capital costs for chilled water storage AC system operating with full strategy

Appendix H

Life Cycle Costs Tables

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	637,838	Base Date	Already in Present Value	637,838
Maintenance at 10.33 ($ per kW_{t})	4,359	25	32	139,748
Annual Electricity Consumption 0.069 ($ per $kW_{e}h$)	37,114	Annual	17	640,499
Annual Water Consumption 2.214 ($ per m^3)	2	Annual	17	37
Total Life Cycle				1,418,121

Table H.1 Estimated life cycle cost of conventional AC system

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	554,196	Base Date	Already in Present Value	554,196
Maintenance at 10.33 ($ per kW_{t})	3,461	25	32	110,947
Annual Electricity Consumption 0.069 ($ per $kW_{e}h$)	37,060	Annual	17	639,527
Annual Water Consumption 2.214 ($ per m^3)	2	Annual	17	38
Total Life Cycle				1,299,826

Table H.2 Estimated life cycle cost of ice storage AC system operating with load levelling strategy

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	640,239	Base Date	Already in Present Value	640,239
Maintenance at 10.33 ($ per kW_{t})	4,004	25	32	128,351
Annual Electricity Consumption 0.069 ($ per kW_eh)	41,201	Annual	17	711,031
Annual Water Consumption 2.214 ($ per m^3)	3	Annual	17	49
Total Life Cycle				1,479,670

Table H.3 Estimated life cycle cost of ice storage AC system operating with 50% demand limiting strategy

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	721,570	Base Date	Already in Present Value	721,570
Maintenance at 10.33 ($ per kW_{t})	4,852	25	32	155,544
Annual Electricity Consumption 0.069 ($ per kW_eh)	44,266	Annual	17	763,926
Annual Water Consumption 2.214 ($ per m^3)	4	Annual	17	62
Total Life Cycle				1,641,101

Table H.4 Estimated life cycle cost of ice storage AC system operating with full strategy AC system

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	508,715	Base Date	Already in Present Value	508,715
Maintenance at 10.33 ($ per kW_{t})	2,556	25	32	81,941
Annual Electricity Consumption 0.069 ($ per $kW_{e}h$)	34,773	Annual	17	600,102
Annual Water Consumption 2.214 ($ per m^3)	42	Annual	17	726
Total Life Cycle				1,191,485

Table H.5 Estimated life cycle cost of chilled water storage AC system operating with load levelling strategy

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	572,985	Base Date	Already in Present Value	572,985
Maintenance at 10.33 ($ per kW_{t})	3,517	25	32	112,760
Annual Electricity Consumption 0.069 ($ per $kW_{e}h$)	34,550	Annual	17	596,244
Annual Water Consumption 2.214 ($ per m^3)	64	Annual	17	1,099
Total Life Cycle				1,283,089

Table H.6 Estimated life cycle cost of chilled water storage AC system operating with 50% demand limiting strategy

Cost Item	Base Date Cost ($)	Year of Occurrence (Year)	Uniform Present Value factor	Present Value ($)
AC Capital	631,234	Base Date	Already in Present Value	631,234
Maintenance at 10.33 ($ per $kW_{t)}$	3,778	25	32	121,099
Annual Electricity Consumption 0.069 ($ per kW_eh)	35,432	Annual	17	611,463
Annual Water Consumption 2.214 ($ per m^3)	81	Annual	17	1,395
Total Life Cycle				1,365,191

Table H.7 Estimated life cycle cost of chilled water storage AC system operating with full strategy

Appendix I

Control System

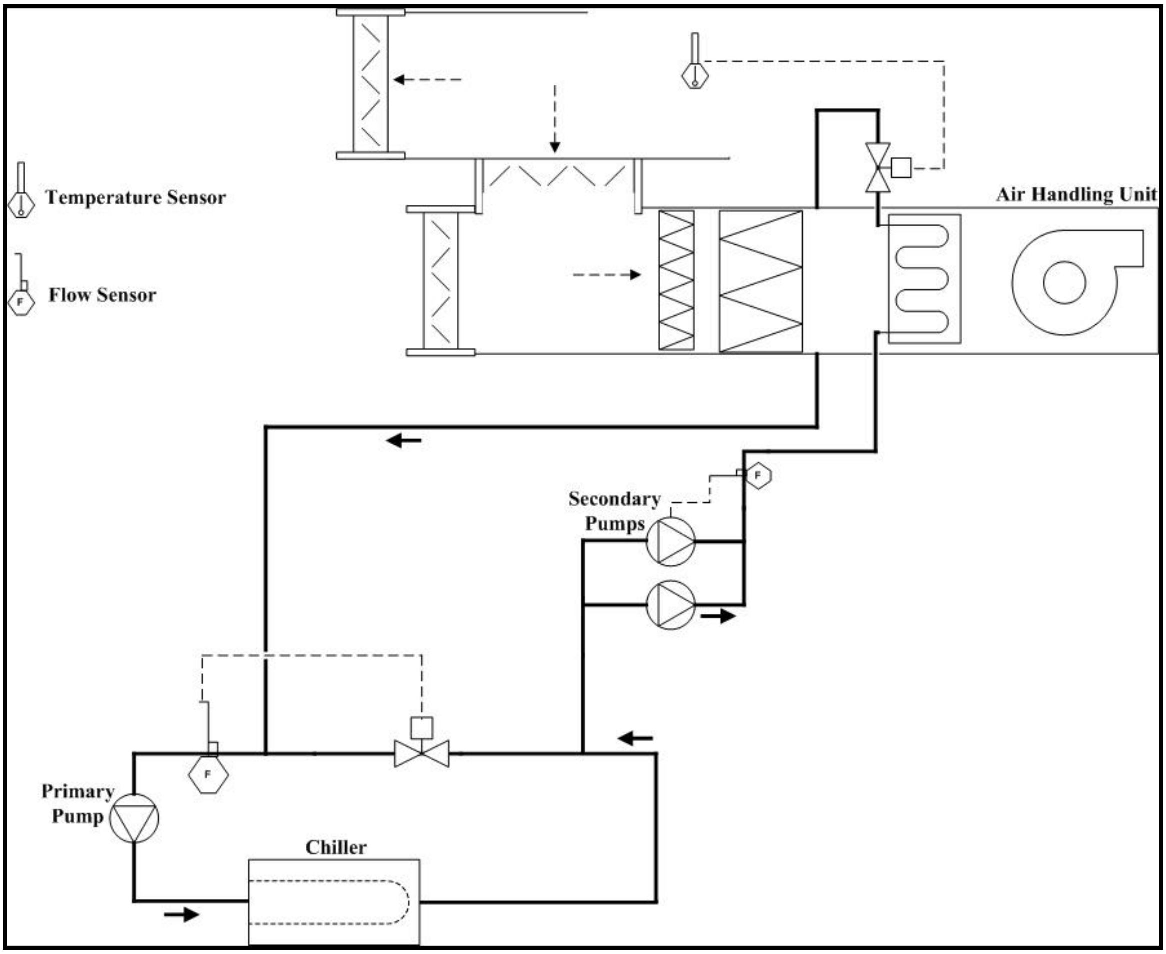

Figure I.1 Conventional AC system.

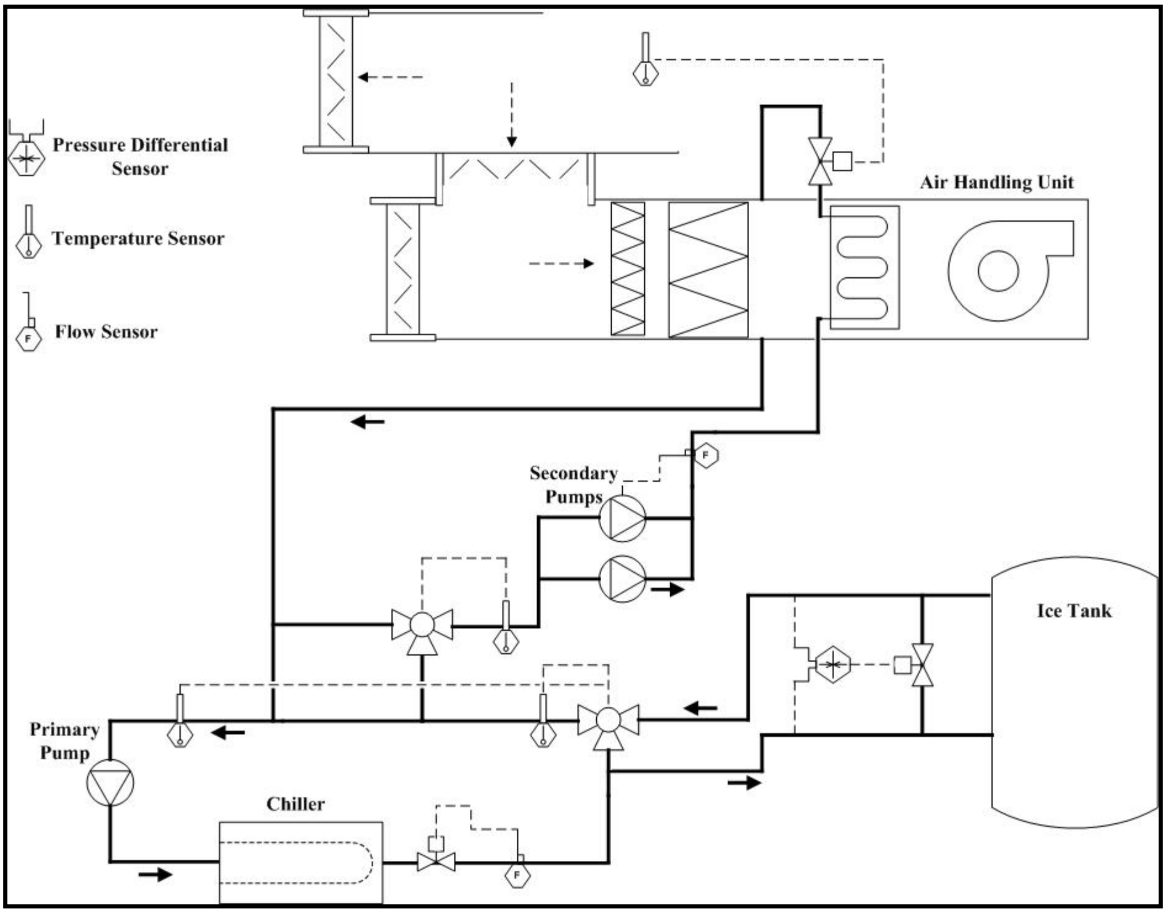

Figure I.2 Ice storage AC system.

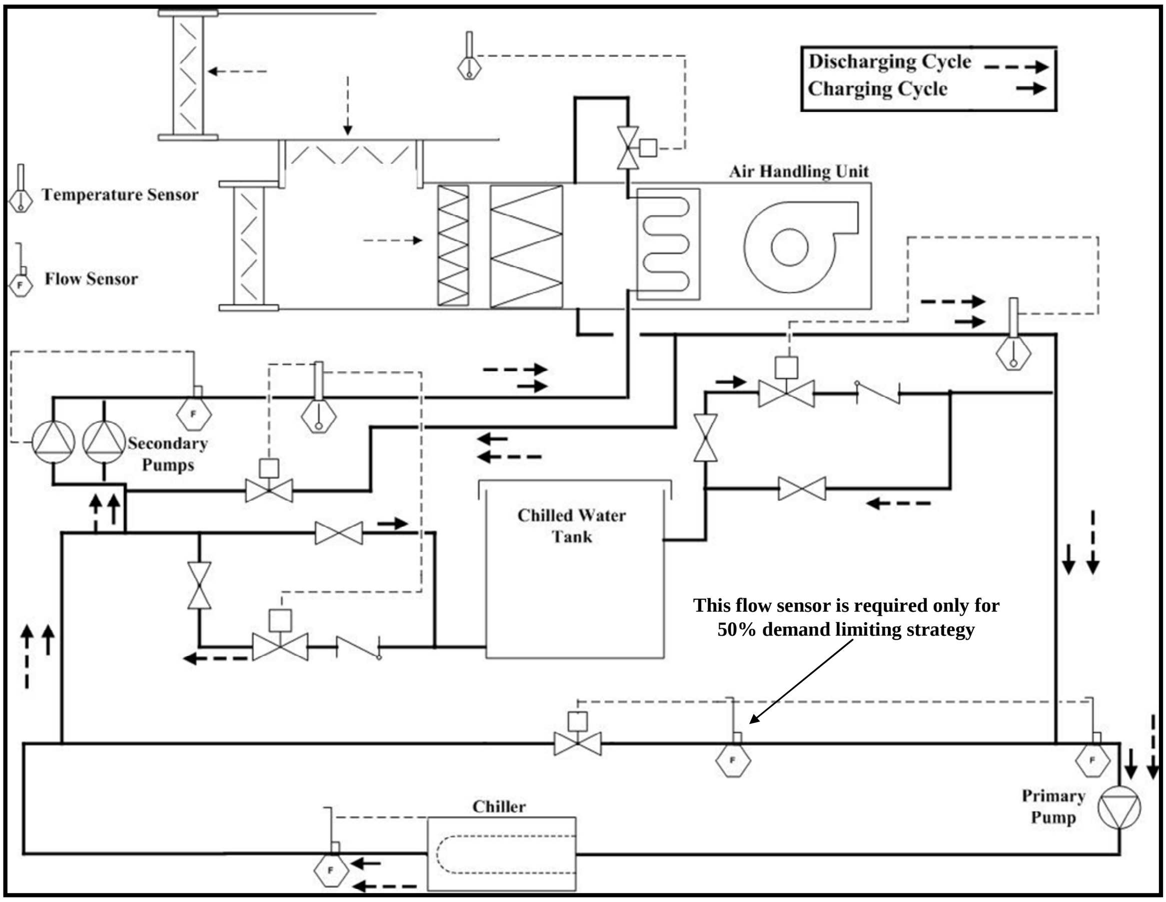

Figure I.3 Chilled water storage AC system.